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CRYOGENIC REFRIGERATORS FOR SHIPBOARD FORWARD LOOKING INFRARED APPLICATIONS

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Cryogenics Division
Institute for Basic Standards
National Bureau of Standards
Boulder, Colorado 80302

Final Report

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U.S. DEPARTMENT OF COMMERCE, Frederick B. Dent, Secretary

NATIONAL BUREAU OF STANDARDS, Richard W. Roberts, Director

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CRYOGENIC REFRIGERATORS FOR SHIPBOARD FORWARD LOOKING INFRARED APPLICATIONS

Roland O. Voth

The Naval Ordnance Laboratory (NOL) has asked the Cryogenics Division of the National Bureau of Standards to investigate and evaluate 1) commercially available refrigerators, 2) refrigerators under development, and 3) new or novel ideas applicable to a refrigerator to cool infrared detectors in a shipboard Forward Looking Infrared (FLIR) system. Although a refrigerator has been selected for two prototype FLIR units, the study was initiated to select the most appropriate refrigerator for additional purchases of FLIR units. The FLIR requires a refrigerator capacity of approximately 2 watts at 77 K and a physical configuration allowing for an interface to the FLIR unit. Information was collected by interviewing FLIR manufacturers, surveying refrigerator manufacturers and by contacting users of similar systems. Correlation of this information with the NOL requirements is presented herein. The primary difference between airborne/spaceborne refrigerators and a shipborne refrigerator is the accessibility for minor repairs on-board ship although major repairs may be deferred for extended periods of time. It is anticipated that the shipboard units will operate away from major maintenance facilities, for periods as long as 6 months, with the refrigerator operating at least half of this time. Thus, reliability and ease of maintenance are emphasized when evaluating the various systems.

Key words: Cryogenics; infrared detector; 77 K refrigerator; low capacity; reliability; shipboard.

1. Introduction

Forward Looking Infrared (FLIR) sensor technology has advanced significantly in the past few years. Since FLIR's can be used at night as well as during the day, they are likely to become indispensable components in future aircraft, ships, missiles, air defense systems, and etc. Currently, great emphasis is being placed on reducing size and weight and developing modular systems which will reduce the cost while improving system reliability and maintainability.

The reliability of early FLIR systems was rather disappointing because they displayed MTBF's (mean-time-between-failures) as low

as 10 hours and frequently less than 100 hours. More recent devices have exhibited higher MTBF values, but these increased values are still not considered satisfactory. Almost invariably, the most unreliable FLIR component has been the miniature cryogenic refrigerator used to cool the multi-element IR detector to cryogenic temperatures. Typical field performance MTBF values for these cryogenic refrigerators lie in the range of 200 to 500 hours. Because (1) experience has shown that FLIR units aboard ship frequently operate 24 hours a day, (2) electro-optical maintenance and repair facilities are practically nonexistent aboard a destroyer, and (3) a ship may be required to sail for several weeks or months before gaining access to repair and maintenance depots, the past reliability of cryogenic refrigerators for use aboard ship is unsatisfactory.

In anticipation that in the future the surface navy will employ a multitude of FLIR devices, the Naval Ordnance Laboratory (NOL) initiated a study to select an "optimum" refrigerator with emphasis on reliability for shipboard use. Reliability, efficiency, size, weight, vibration spectra, manufacturing and interfacing considerations, and cost were included during the selection process.

The study was approached by (1) determining the required characteristics for the cryogenic refrigerator by talking to FLIR manufacturers; (2) determining the characteristics of currently available refrigerators and of refrigerators being developed under the sponsorship of the U. S. Army, U. S. Air Force, and NASA as they relate to the projected shipboard requirements; (3) performing a literature survey for information pertaining to low capacity refrigerators; (4) performing a trade-off analysis to determine the most suitable cryogenic device for adoption into the shipboard FLIR unit;

and (5) determining if any new or novel ideas could be incorporated into the cryogenic refrigerator to increase reliability or other desired characteristics.

2. Cycles and Definitions

In order to select a refrigerator for a particular application, it is useful to understand the various thermodynamic cycles used in cryogenic refrigerators and to understand the various inherent qualities of each cycle. As is normal in thermodynamics, the efficiency of a thermal device is compared to an ideal cycle. The ideal cycle used to evaluate cryogenic refrigerators is a Carnot cycle. Figure 1 is a schematic representation of a Carnot refrigerator together with a temperature-entropy representation of the state of the fluid within the cycle. The Carnot refrigerator removes heat from a low temperature source and transfers this heat to a higher temperature sink using input energy consisting of mechanical work. The Carnot cycle consists of four processes: (1) An isothermal reversible process to remove the heat, Q_2, from the refrigerator, (2) a reversible adiabatic process in which the temperature of the fluid drops from T_2 to T_1, (3) a constant temperature process in which heat, Q_1, enters the refrigerator, and (4) the reversible adiabatic process increasing the fluid temperature from T_1 to T_2. The quantity of heat rejected, Q_2, must equal all energy inputs to the system, or

$$Q_2 = Q_1 + W_c ,$$

where W_c is the input work, usually the work of compression. In refrigerators operating at cryogenic temperatures (77 K), the quantity

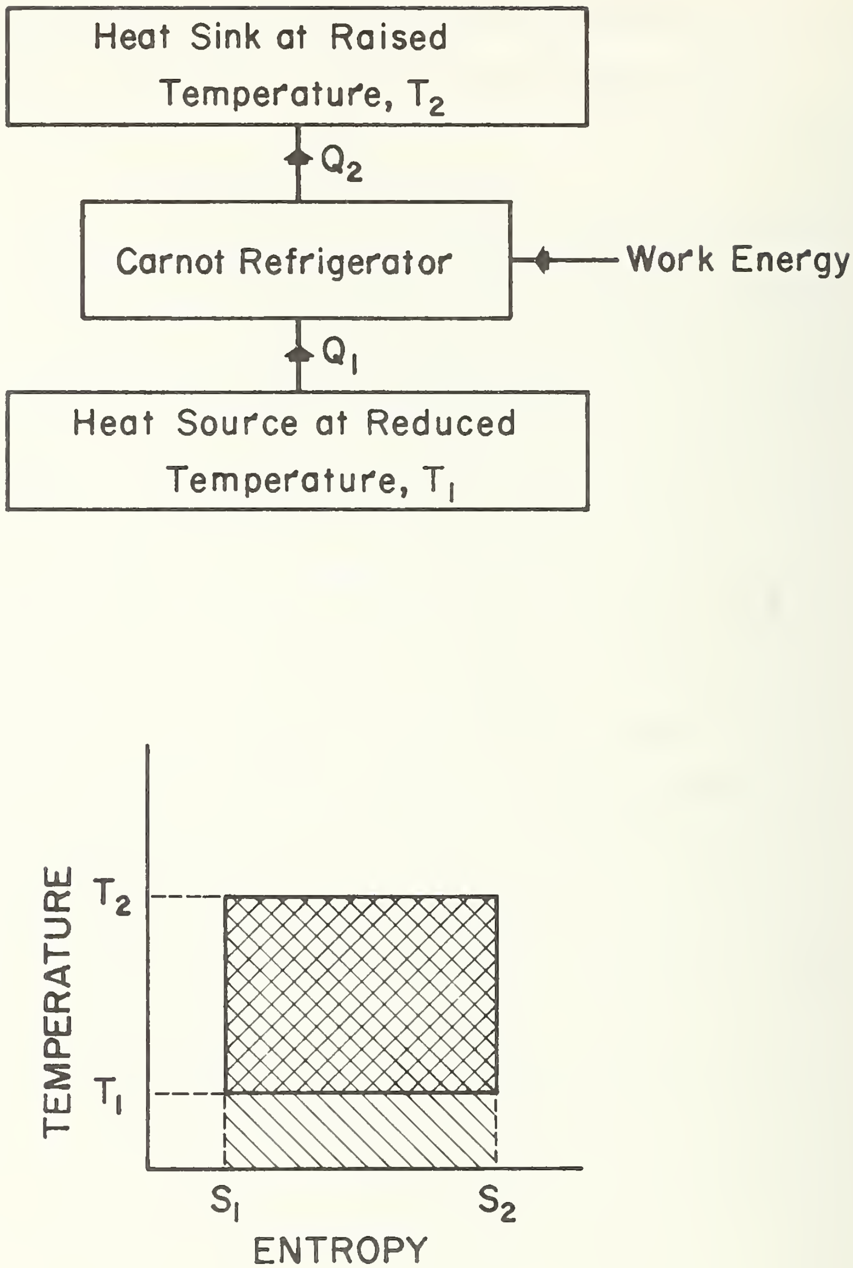


Figure 1. Schematic of a Carnot cycle refrigerator.

of heat, Q_1, is small in relationship to the work term, W_c, and is usually ignored when calculating the heat rejected from the system.

From the temperature-entropy diagram (fig. 1), the net amount of work input into the Carnot cycle is equal to the double crosshatched area, or

$$W_c = T_2 (S_2 - S_1) - T_1 (S_2 - S_1) . \tag{1}$$

The useful refrigeration is equal to the single crosshatched area, or

$$Q_1 = T_1 (S_2 - S_1) . \tag{2}$$

Thus, the ideal input work per unit refrigeration for a Carnot refrigerator is equal to

$$\frac{W_c}{Q_1} = \frac{T_2 - T_1}{T_1} . \tag{3}$$

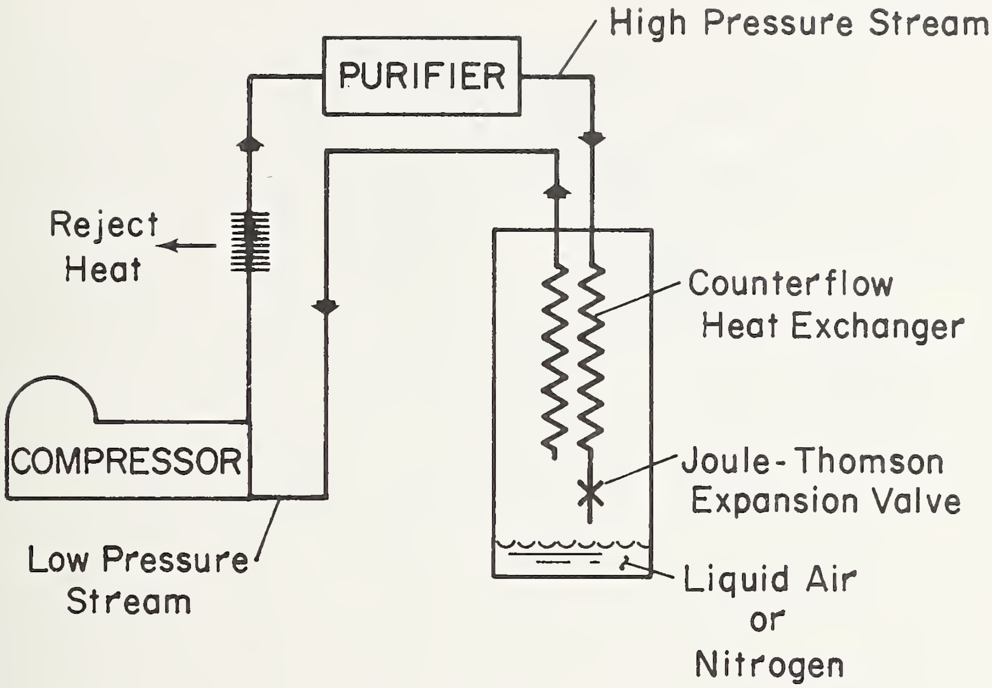
The reciprocal of W_c/Q_1 is the Coefficient of Performance (COP) of an ideal Carnot refrigerator. Efficiency of a refrigerator will be defined as the ideal work per unit refrigeration divided by the work per unit refrigeration for an actual system,

$$\text{Eff} = (W_c/Q_1)_i / (W_c/Q_1)_a . \tag{4}$$

All gaseous refrigeration cycles use one or both of two basic processes to obtain cooling. The first process is the expansion of gas in an increasing volume, as above a piston, where work is removed from the fluid during the expansion process, resulting in a temperature

reduction. The second process depends on a nonideal gas where even though the enthalpy of the gas remains constant during a reduction in pressure due to throttling, the temperature of the gas decreases. The second process is called the Joule-Thomson expansion process.

A schematic of a closed cycle Joule-Thomson (J-T) refrigerator is shown on figure 2. In the system, a mechanical compressor increases the pressure of flowing gaseous nitrogen. After the removal of the heat of compression and remaining gas impurities, the high pressure gas is cooled by returning low pressure, low temperature gas in the counterflow heat exchanger. The gas is then expanded through the J-T valve, further reducing its temperature, and after removing heat from the load, returns to the compressor through the returning leg of the heat exchanger. Because the J-T expansion effect depends on the gas being nonideal, high pressure levels are required in the cycle. The high pressure required for nitrogen gas ranges between 12.2 MPa(120 atm) and 40.5 MPa(400 atm) when entering the J-T heat exchanger. Two types of J-T cryostats are available for use with infrared detectors. The first type uses either a small orifice of a fixed size at the high pressure discharge end of the counterflow heat exchanger or a capillary tube for the high pressure side of the heat exchanger to provide the required pressure drop. The capillary tube has a larger cross section area than the orifice for an equal capacity refrigerator, thereby reducing the inherent plugging problems of the orifice. The second type of J-T cryostat is termed a "demand flow" cryostat. The "demand flow" cryostat has an orifice with a variable cross section area. The area of the orifice is dependent on the refrigeration temperature and is



The compressor and purifier of the Joule-Thomson refrigerator can be split or located remotely from the cold head.

Figure 2. Schematic of a closed cycle Joule-Thomson refrigerator.

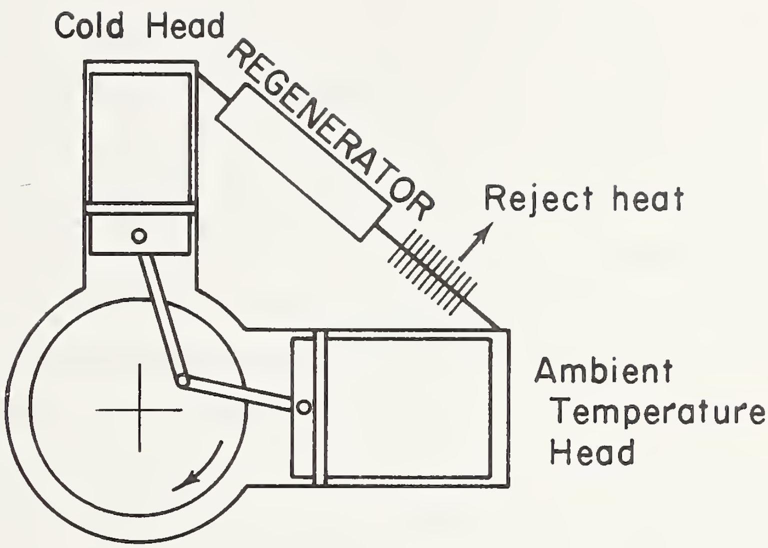
controlled by a small nitrogen-filled bellows attached to the cold end of the heat exchanger. The "demand flow" cryostat is able to adjust the gas flow rate as the required refrigeration capacity or supply pressure changes.

The "demand flow" cryostat is generally used in open cycle applications while the fixed area cryostat is used in closed cycle applications with a fixed capacity compressor. A more technical discussion of Joule-Thomson refrigerators can be obtained from [1].

The Stirling cycle refrigerator uses work-removing fluid expansion to obtain reduced temperatures. With correct design, the path of the cycle on a temperature-entropy diagram nearly duplicates a Carnot cycle. Thus, the Stirling cycle is a relatively efficient cycle, and because it is efficient, the unit is usually physically smaller than any other refrigerator of comparable capacity.

The Stirling cycle employs regenerators which are constructed of low thermal conductivity enclosures containing lead balls or stacked perforated disks. During steady state operation, a temperature gradient exists along the length of the regenerator such that warm gas passing through the regenerator is cooled by adding heat to the regenerator, and under reverse flow conditions heat is removed from the regenerator to warm the flowing gas. Ideally, the regenerator should have a high specific heat, extended heat transfer area, a low void volume, low pressure drop, and a low longitudinal thermal conductivity. A more detailed discussion of regenerators can be found in [2].

Figure 3 is a sketch of a Stirling cycle refrigerator. Visualizing rotation of the crankshaft, the working volume, which consists of the volume above the pistons and in the regenerator, is changing. This changing volume causes pressure variation in the working volume.



The Stirling cycle refrigerator is an integral machine.

Figure 3. Sketch of a Stirling cycle refrigerator.

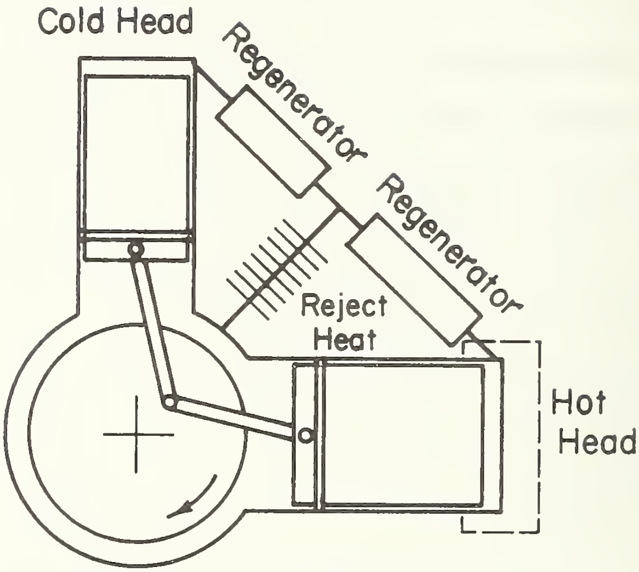
The location of the majority of gas in the system also changes with shaft rotation. At high pressure, with the warm piston at top dead center, most of the gas is in the low temperature cylinder. As rotation continues from top dead center; the gas in the cold cylinder is expanded (reducing its temperature) by the cold piston going down and by gas leaving the cold area and returning to the warm cylinder. The gas flowing to the warm cylinder is warmed during its passage through the regenerator. Finishing the cycle, the gas again flows to the cold piston after being cooled by the regenerator. A more technical discussion of Stirling cycle operation can be obtained from [3].

Low capacity Stirling cycle refrigerators are available to cool infrared detectors. Their major advantages are small size and high efficiency while major disadvantages are reduction in performance with long term operation, high vibration loads imparted to any connected loads, and low MTBF values for currently available machines. The degradation in performance is due to contamination of the relatively low volume of working fluid (usually helium) with lubrication materials from the crankcase and other outgassed materials from components such as electric drive motors that are usually sealed into the crankcase area. The vibration loads are from the reciprocating pistons. Stirling machines can be developed to improve or eliminate these disadvantages. A rhombic drive to eliminate or reduce vibrations and a positive rolling seal between the working volume and the crankcase to eliminate contamination can be utilized to improve the machine. The positive seals also allow the use of oil-lubricated bearings in the crankcase instead of dry-lubricated bearings, significantly increasing the life of the unit.

The Vuilleumier (VM) cycle refrigerator bears a resemblance to the Stirling cycle units with one major difference; the crankcase

volume is connected to the displacer volumes, making the working volume of the refrigerator constant during rotation of the crankshaft. Varying pressure with crankshaft rotation is accomplished by heating the high temperature cylinder to approximately 920 K. Although the working volume of the cycle is constant, the average temperature and therefore the average pressure of the working gas varies as the crankshaft rotates. Figure 4 is a sketch of a VM refrigerator. When the hot displacer is at the bottom of its stroke, the average temperature of the gas in the working volume is high, resulting in a high pressure, and as the crankshaft continues to rotate, the pressure is low when the hot displacer is at top dead center. Expansion of the gas occurs during this rotation. The motor used to rotate the crankshaft of a VM refrigerator is small because it only needs to overcome sliding friction and pressure drops associated with moving the gas inside the machine. The majority of the input power is introduced at the hot cylinder making the VM refrigerator particularly attractive when a high temperature heat source is already available. For the FLIR application, an electric heater would be used to heat the high temperature cylinder. A more technical description of the Vuilleumier cycle can be obtained in [4, 5, and 6].

The Vuilleumier (VM) machine has the small size advantage of the Stirling cycle refrigerator with additional features such as low loading on the bearings and seals. The low loading allows the use of lubrication and bearing materials less likely to contaminate the helium working fluid. Also, the torque required to operate the refrigerator is low, allowing a different design of the drive motor to eliminate contaminates from this source. Because only a low pressure differential exists across the displacers, they can be of lighter construction than a Stirling cycle, thus reducing vibrations due to the reciprocating motion of the

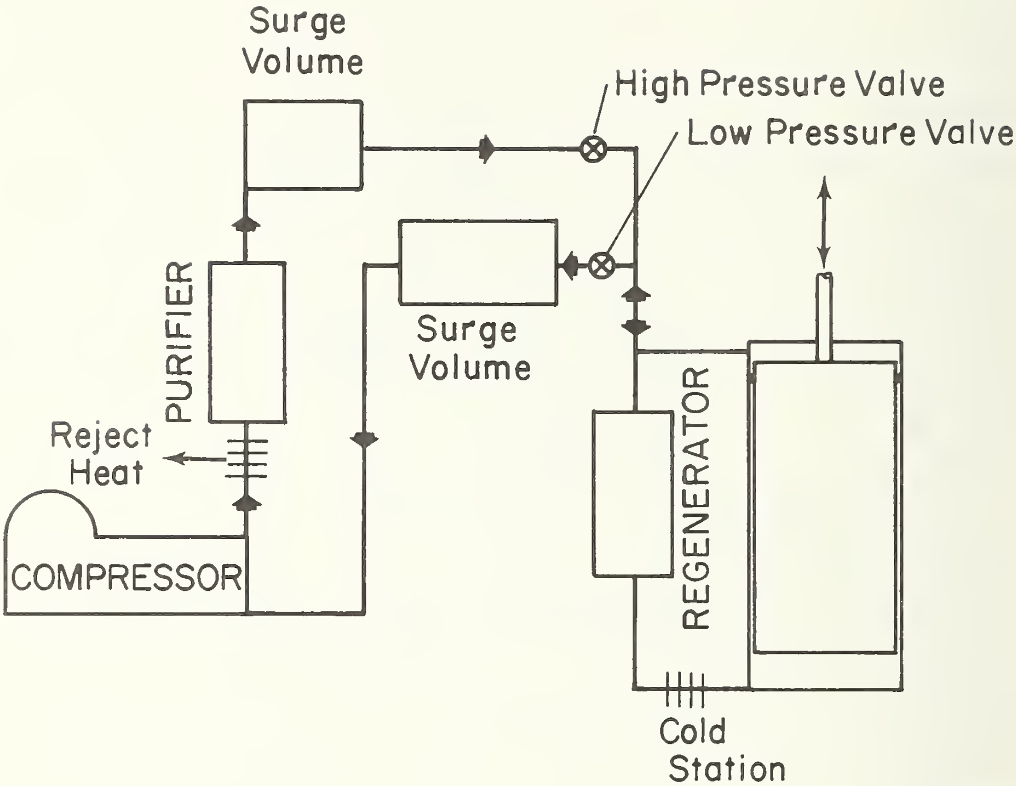


The Vuilleumier refrigerator is an integral machine.

Figure 4. Sketch of a Vuilleumier refrigerator.

displacer. Major disadvantages of the cycle include the complex electronics to control the motor speed and the power to the electric heater on the hot cylinder. Both of these problems can be solved using state-of-the-art technology. The high temperature cylinder design has been a problem which appears nearly solved. Experimentation to increase the life of the VM machine and decrease its cost is still progressing. The internal helium pressure is quite high, especially when the refrigerator is at ambient temperature, making helium leakage from the working volume a problem aggravated by extremes in the environment. The external seals are being developed, however, and the problem of helium leakage should diminish with future machines.

One refrigerator related to the Stirling cycle is the Gifford-McMahon cycle. This cycle has some of the same inherent qualities of a Stirling and VM cycle such as reversing flow in regenerators; also it has better reliability because an inline gas purifier is used to remove impurities from unidirectional gas flow before it enters the low temperature portion of the refrigerator. Figure 5 schematically shows a single-stage Gifford-McMahon unit. The low thermal conductivity cylinder is fitted with a displacer which is sealed against the warm part of the cylinder. The displacer movement is controlled from outside the cryostat. A thermal regenerator is also shown in the circuit. Imagine the displacer in its lowest position with steady-state conditions existing so that proper thermal gradients exist in the thermal regenerator. When the compressed-gas supply valve is opened, helium enters the system, raising the pressure both in the volume at the top of the cylinder and throughout the system. When the pressure in the regenerator and in the volume above the displacer reaches the maximum,



The compressor, purifier and surge volumes can be split or located remotely from the cold head.

Figure 5. Schematic of a Gifford-McMahon cycle refrigerator.

the displacer is moved to the upper position, forcing the gas above it through the regenerator into the volume which is now below the displacer. The gas is cooled as it gives up heat to the cold regenerator, decreasing its specific volume and allowing more helium into the system from the compressed gas supply. The supply valve is closed, and the exhaust valve is slowly opened. Refrigeration is now produced as each element of gas does work on elements of gas being exhausted. The fluid is warmed as it passes through the regenerator and through the heat absorption station on its way to the compressor. When the pressure in the system has reached its lowest level, the displacer is moved to its lowest level, forcing the remaining cold gas into the compressor intake. At this time, the original thermal pressure state is achieved and the cycle repeats. A more technical description of a Gifford McMahon and related cycles is in [2].

Several variations of the split cycle exist. These variations carry different names such as Split Stirling Cycle, Modified Stirling Cycle, Split Vuilleumier Cycle and modified Solvay Cycle. Most of these cycles have a free piston expander. Figure 6a shows a cross section of the free expander portion of such a cycle. The free piston works if a pulsating pressure, ambient temperature gas is supplied to the inlet tube. The seal around the small piston into cavity A is designed to allow a low rate of gas leakage from above the displacer into the cavity. This low leakage rate maintains the pressure in cavity A about halfway between the maximum and minimum pressures in the displacer cavity. Thus, as the pressure in the displacer cavity is increased, high pressure works over a larger area on the bottom of the displacer pushing it up. As the pressure in the displacer cavity decreases to a low pressure, the remaining pressure in cavity A pushes the displacer back to the

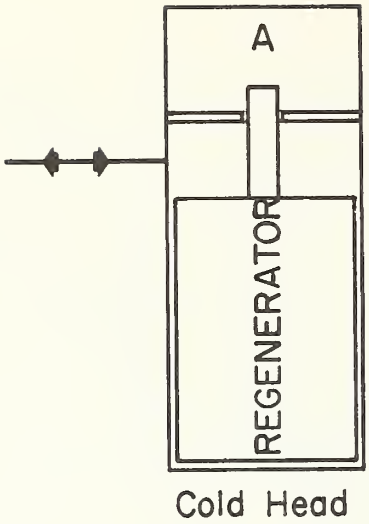


Figure 6a. Free expander

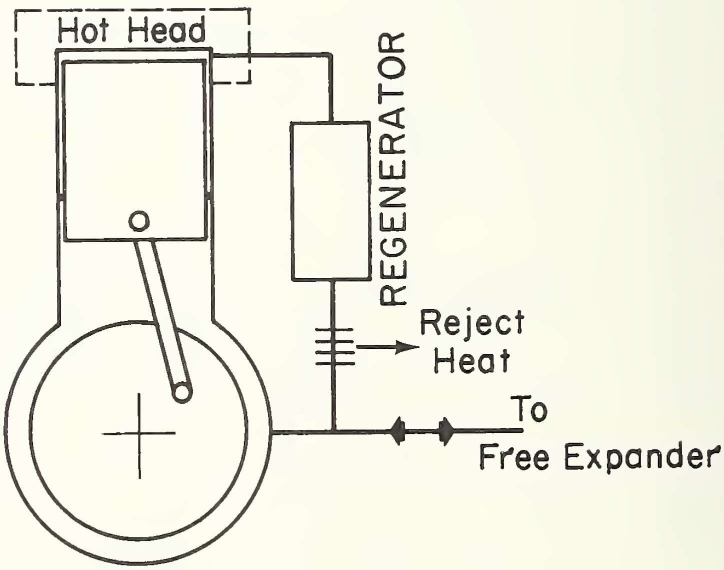


Figure 6b. Thermal compressor.

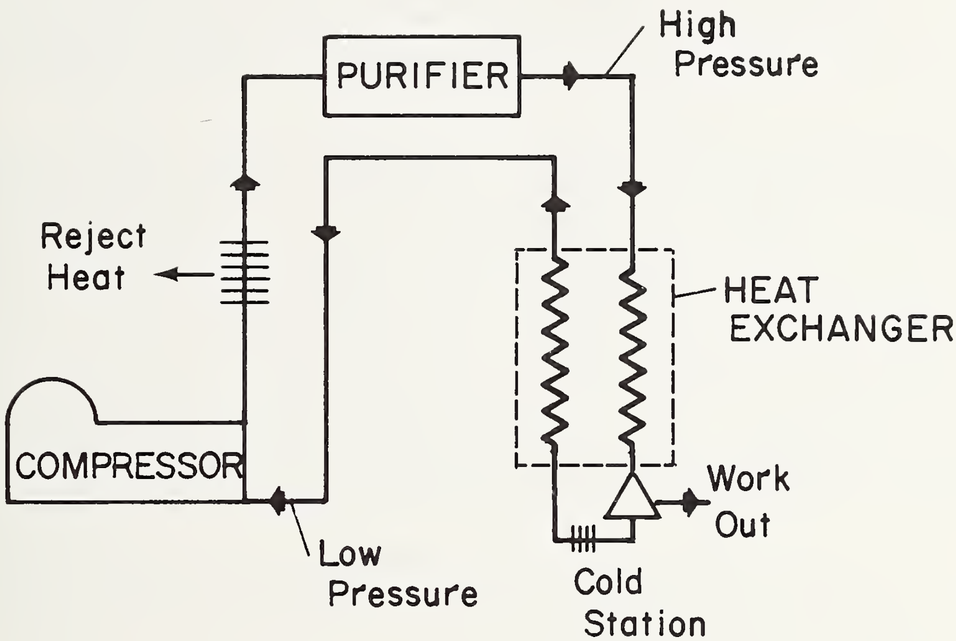
Figure 6. Sketches of a free expander and a thermal compressor.

lower position. The different names applied to the free piston system reflect the type of compressor used with the system. In some instances, the varying pressure is supplied by a valveless reciprocating compressor connected to the expander by a single tube. As the piston in the compressor moves, the volume of the system varies, resulting in a pulsating pressure. This cycle is termed a modified Stirling cycle or split Stirling cycle. If the pulsating pressure is supplied by a thermal compressor (figure 6b), the cycle is termed a split VM cycle. In this case, the pressure in the cycle is again determined by the average gas temperature in the system. This cycle offers some of the same advantages as the integral VM refrigerator with a low torque motor requirement together with low stress bearings and seals in the compressor. The free piston expander can also be used with a compressor with external surge volumes and valves to obtain the pulsating pressure. This type of connection (similar to the compressor connections shown in figure 5) alternately connects the free expander to a high pressure surge volume or to a low pressure surge volume. This cycle is termed the modified Solvay cycle. The advantage of the free expander is the single tube interconnection between the compressor system and the cold head. Also, no electrical power is required by the expander, reducing EMI problems, and if the expander is cycled at high speed, it can be physically very small, allowing interchangeability with the Joule-Thomson cryostat. Cne major difficulty of the free piston expander when used with the thermal compressor is the volume of the interconnecting line. This line must be kept fairly short in order to maintain a low volume of gas in the line so that the refrigerator performance is not materially affected.

Figure 7 is a schematic of a reversed Brayton cycle refrigerator. The Brayton cycle refrigerator appears similar to the J-T refrigerator except that a work producing expander is substituted for the J-T valve. Refrigerator efficiency is increased by using a work extracting expander and the pressure levels necessary for efficient operation are lower than the J-T system. The Brayton cycle is used extensively for larger refrigerators; however, in low capacity units the expansion device becomes a design problem. Low capacity reciprocating expanders are relatively short lived, and efficient turbo expanders are difficult to manufacture because of the low volume flow rates encountered in low capacity units. Like the J-T system, gas contamination is a problem because unlike a regenerator where reversing flow tends to purge contaminates from the regenerator, the flow through the heat exchanger, is always in the same direction, tending to continually freeze out contaminates at the same location in the heat exchanger. During extended runs, the heat exchanger will tend to become plugged by even minute quantities of foreign materials in the circulating fluid. Because of these plugging characteristics and the expander engine design problems, the use of low capacity Brayton cycle refrigerators has been limited to developmental machines for space applications where the desirable characteristics override the other problems.

3. Operational Requirements

The operational requirements for the cryogenic refrigerator are dependent to a large extent on the design and location of the FLIR unit. Both the overall FLIR design and its location onboard ship were considered a variable during the refrigerator evaluation. The multi-element detectors are mounted within an enclosure supported by double gimbals, allowing them to "see" in any selected direction. Mirrors and lenses focus the incoming radiation onto the detectors, cooled to around 77 K by a cryogenic refrigerator. Electronics associated with



The compressor and purifier of the Brayton cycle refrigerator can be split or located remotely from the cold head.

Figure 7. Schematic of a Brayton cycle refrigerator.

the detectors are also within the enclosure, making the EMI emitted by the refrigerator a major problem which can be eliminated with proper shielding. Refrigerators or the refrigerator cold head attached to the detectors must be capable of operation in any orientation as the FLIR unit is constructed to allow it to look horizontally and straight up anywhere in a 360 degree arc. Currently the rotation is limited to 360 degrees. The compressor for a split cycle refrigerator can be mounted on the outer gimbal or on the deck. This type of mounting limits the variance of the compressor from horizontal to the roll or pitch of the ship. Initial FLIR units will be deck mounted, allowing accessibility for maintenance, but subsequent units may be mounted in the superstructure of the ship to give a view less impaired by the local surroundings while increasing the range of observations. For deck mounted units, the size of a split cycle compressor is not critical; however, for units mounted high on the superstructure, large size and weight become a liability.

The environment in which the refrigerator must operate will closely follow the ambient environment surrounding the ship except, in some instances, where the unit mounted in the superstructure may be affected by stack gases. Thus, the environmental conditions surrounding the refrigerator will be approximated by the extremes of the earth's climate.

3.1 Cryogenic Requirements

3.1.1 Refrigeration Temperatures and Temperature Stability

In evaluating a refrigerator for use in a FLIR system, it is essential to consider, among other things, how the refrigeration temperature will affect (a) the thermal sensitivity, (b) the spectral transfer function, and (c) the modulation transfer function of the FLIR. These parameters are in turn related to the detectivity, responsivity, noise spectra, spectral response, and frequency response of the detector array to be cooled.

In the present study, attention is focused on arrays of $Hg_{1-x}Cd_xTe$, which are intrinsic photoconductive infrared detectors. The value of x can vary considerably, resulting in various spectral responses and other characteristics. However, in the FLIR systems of interest, the detectors used have $x \approx 0.2$, resulting in the typical spectral response shown in figure 8 for detector temperature equal to about 77 K.

Considering the effect of detector temperature on its responsivity and noise characteristics, it is more convenient to examine the specific detectivity (D*), which incorporates both the responsivity and noise of a detector. Since both thermally and optically generated carriers are noise sources for an infrared detector, both the detector temperature and background photon flux affect the detectivity. The detectivity is also affected by the temporal modulation of the IR radiation impinging upon the detector.

The background photon flux and temporal modulation of the incident radiation are related to the field of view and scanning characteristics of the FLIR and are therefore beyond the scope of the present study. As for the effects of detector temperature upon D*, it has been well established that D* varies typically with temperature as shown in figure 9, where a rather rapid decrease in D* is seen to begin at about 80 K. For temperatures below 80 K, and particularly in the region of 60 to 80 K, the change in D* with temperature is insufficient to cause any perceptible change in the visual display of the FLIR unit. This behavior is attributed to Shockley-Read recombination effects [7]. Exception to this must be taken in the 1/f - noise region, which, however, is of no concern in the case at hand, since all FLIR systems are designed to modulate the incoming radiation well beyond the "knee"

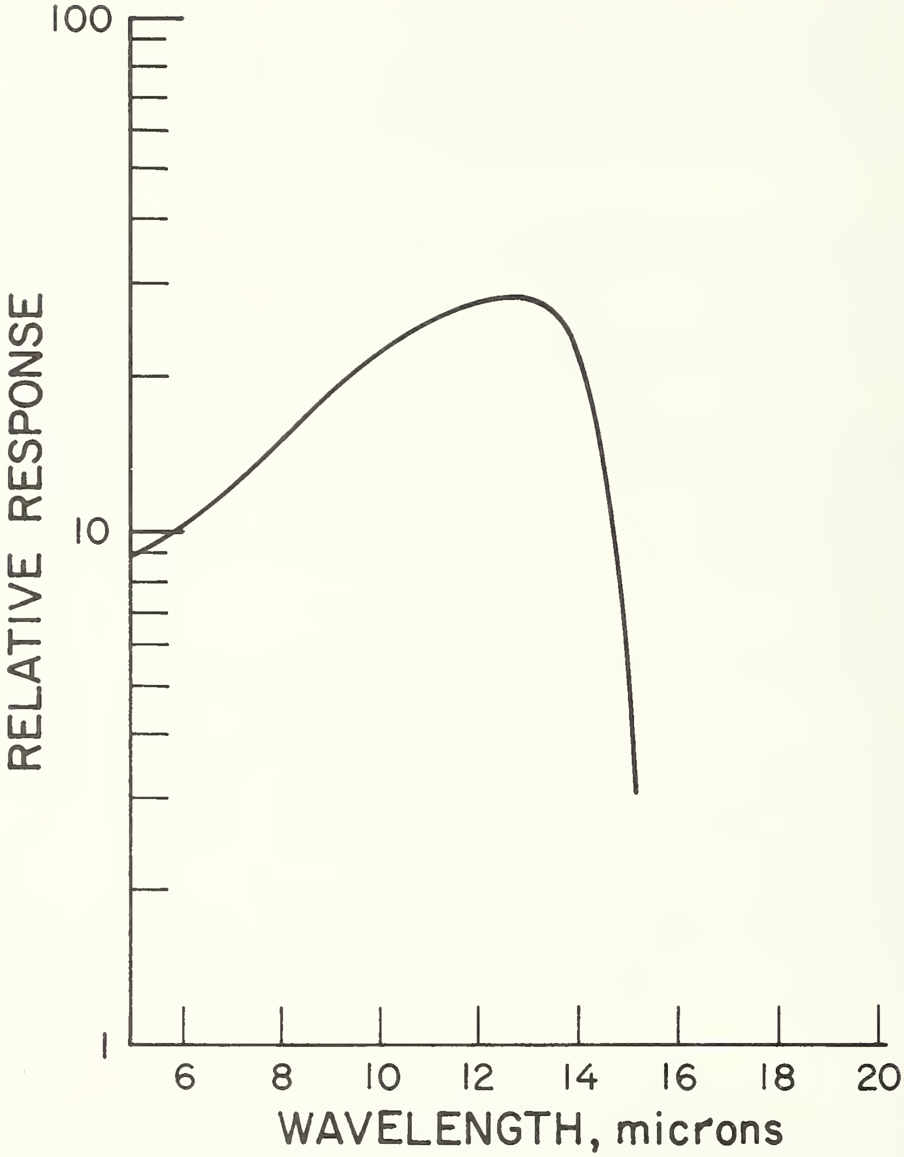


Figure 8. Typical response of a $Hg_{.8}Cd_{.2}Te$ detector.

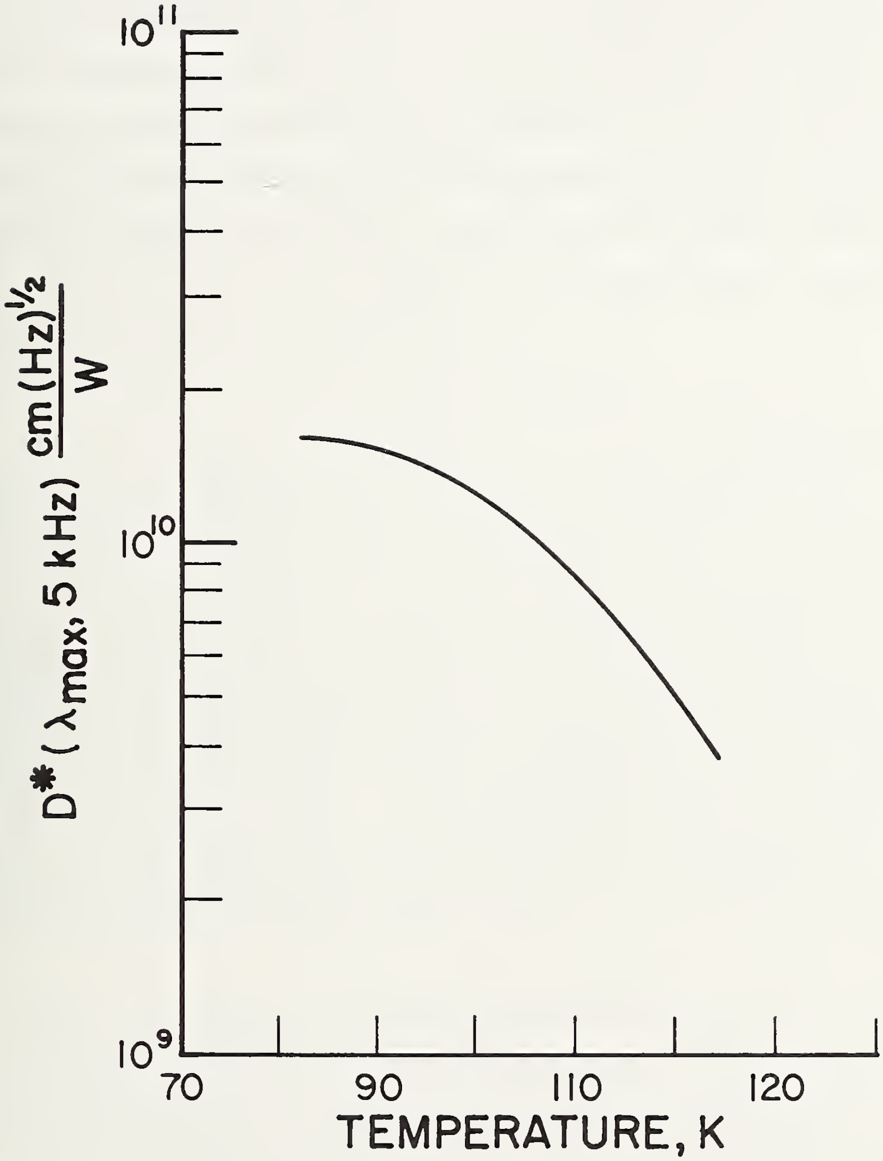


Figure 9. Typical D* of a $Hg_{.8}Cd_{.2}Te$ detector.

in the 1/f - noise region. It can be thus concluded that any refrigerator capable of maintaining a mercury cadmium telluride detector array at any temperature between 60 and 80 K should result in inappreciable change in D* and, consequently, in the thermal sensitivity of the FLIR.

With regard to changes in the detector spectral response due to variations in its temperature, it is a well-known fact that the spectral response shifts to shorter wavelengths as the temperature increases. It has been determined [8] that for $Hg_{1-x}Cd_xTe$ such spectral shifts can be expressed by the empirical relation

$$Eg = 1.59x - 0.25 + 5.233(10^{-4})\,T(1-2.08x) + 0.327x^3, \quad (5)$$

where Eg is the energy gap in eV; Eg can be converted to λ_{co} (long wavelength 50% cutoff) by the relation $\lambda_{co} = \frac{hc}{Eg} = \frac{1.24}{Eg}$. Applying eqs 1 and 2 to $Hg_{1-x}Cd_xTe$ with $x = 0.2$, we find that at $T = 60$ K, $\lambda_{co} = 13.9$ μM, and at $T = 80$ K, $\lambda_{co} = 13$ μM. Thus, in the range of 60 to 80 K, the spectral shift is only 0.9 μM. Since the atmospheric window in the 8 to 14 μM region has an effective wavelength cutoff at about 12.5 μM over long horizontal over-water paths, it can be concluded that changes in detector temperatures in the 60 to 80 K range will not produce any effective changes in the spectral transfer function of a shipboard FLIR.

Finally, concerning variations in detector frequency response with temperature, it has been established that no such changes occur. Therefore, the modulation transfer function of a FLIR should not be affected whether the detectors are cooled to 60 K or 80 K.

3.1.2 Refrigerator Capacity

The study has been concerned with refrigerators capable of providing one to two watts of refrigeration in the temperature range of 60 to 80 K. This refrigeration capacity was chosen because the existing

shipboard FLIR units require between 1.5 and two watts of refrigeration during steady state operation. Future systems are expected to require refrigeration capacities nearer one watt. Because of the expected decrease in refrigeration loads, two evaluations were performed--first, for a two watt capacity and then for a one watt capacity unit. The 60 K to 80 K temperature range was chosen because all shipboard FLIR units, whether existing or under development, use mercury cadmium telluride detectors operated in this range, and it can be safely predicted that future FLIR units will almost exclusively use detector alloys requiring cooling in the neighborhood of 77 K.

3.1.3 Cooldown Time

Cooldown time for a system which is not running continuously is an important operational refrigerator characteristic. A cooldown time of 25 minutes or less was determined to be a requirement for the shipboard FLIR application. Cooldown times for a refrigerator with an attached thermal mass containing 3000 joules of latent heat linearily distributed between 300 to 70 K were used to evaluate refrigerators for current FLIR units. Future FLIR units are expected to decrease the attached thermal mass by a factor of two; therefore, during the evaluation of one watt refrigerators, calculated cooldown times for a 1500 joules thermal mass were used.

3.2 Mechanical Requirements

The mechanical design of FLIR units is not fixed and may change as the mounting location aboard ship is changed and as the state-of-the-art for scanning detectors advances. Thus, the mechanical details of cryogenic refrigerators for use with FLIR units are not fixed. Only opinions of the FLIR manufacturers describing preferred mechanical details can be expressed.

3.2.1 Location of Refrigerators

Figure 10 is an outline sketch of the receiver turret of a prototype FLIR unit being delivered to the NOL for deck mounting. The drawing is shown to an approximate scale of 1 cm = 12 cm. The unit is relatively large when compared to airborne units. The supplied cryogenic refrigerator for the unit is a split cycle unit with the compressor occupying position A on the outer gimbal and the cold head of the refrigerator occupying position C within the detector enclosure. Flexible lines interconnect the two units. Position B shown on the drawing indicates the possible location of a larger compressor, with flexible lines being used to cross two gimbals. Of course, the integral unit Vuilleumier and Stirling cycle refrigerators would be located at position C with only electrical power connections being led through the two gimbals.

The FLIR manufacturers' preferences appeared to be for an integral unit located at position C. Various reasons given for this preference are: (1) flexible gas lines connecting the compressor to the cold head together with the many electrical wires leading to the unit make construction complex; (2) if the interconnecting lines are springy and apply a residual torque through the gimbals, the positioning motors would need to be larger to maintain the gimbals at a selected position; (3) the reliability of the interconnecting flexible tubing, constantly flexing, was questioned. These disadvantages of split cycle units must be weighed against those of integral units which are: (1) the size of an integral unit mounted within the detectors enclosure would generally be larger than a comparable capacity, split unit, cold head; this increased size may require a larger overall detector enclosure; (2) heat rejected by the cryogenic refrigerator would be added to heat generated within the enclosure by other electrical components; cooling complications

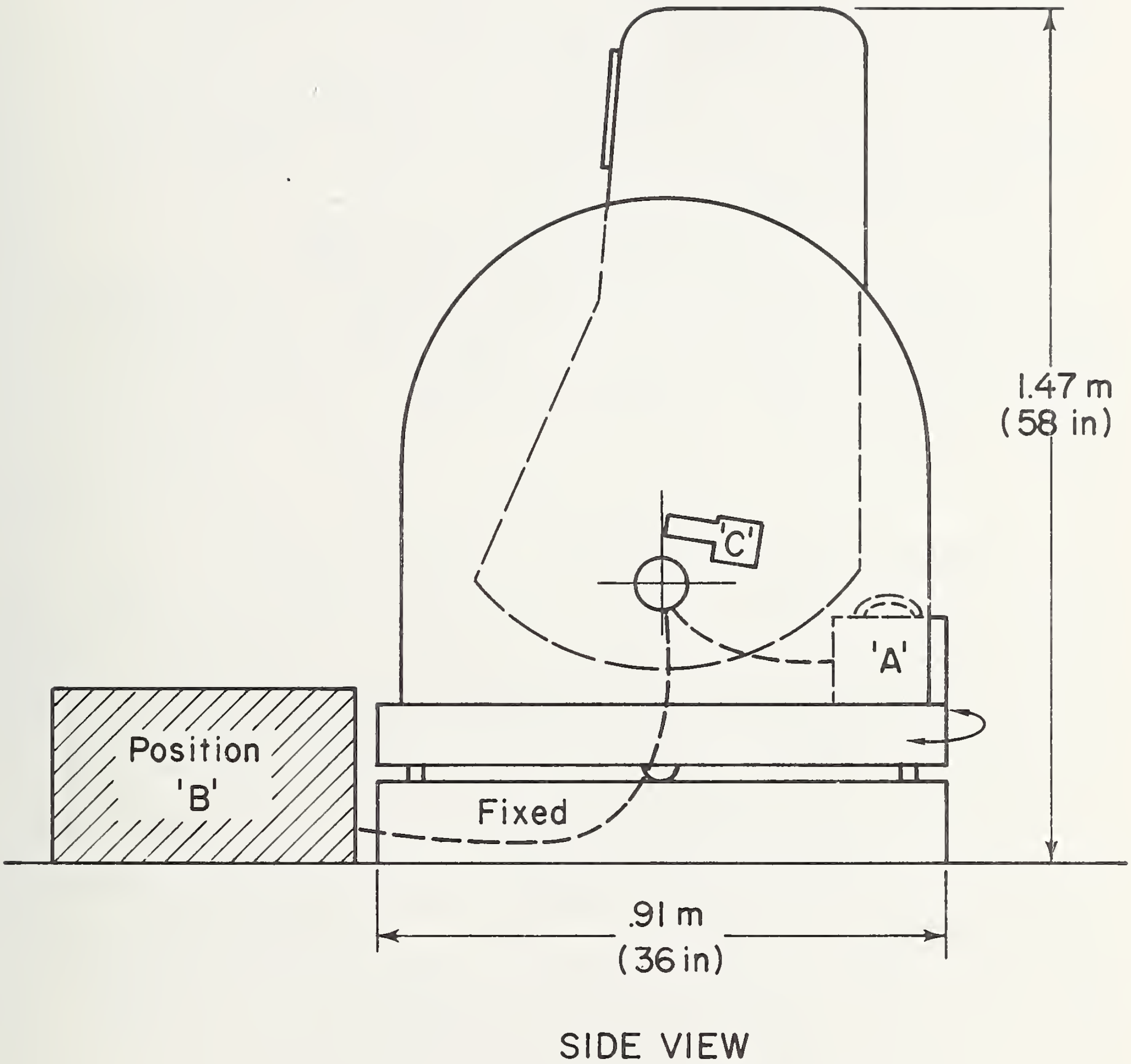


Figure 10. Outline sketch of the receiver turret of a shipboard FLIR unit.

may arise from the additional heat; and (3) accessibility for repair of the integral refrigerator and split cycle cold box would be nearly the same; however, a compressor located outside the enclosure would be more accessible for maintenance.

The conclusion was that split units, especially with large compressors, may not be as desirable as smaller integral units although the differences are not great.

3.2.2 Refrigerator Size and Weight

Historically, the size and weight characteristics of miniature cryogenic components developed for FLIR's have been dictated by constraints imposed by aircraft environments. One of the results of these constraints has been rather poor reliability. On the other hand, the shipboard environment allows for some relaxation regarding size, weight and power consumption. It may also allow, in some cases, the use of other cooling schemes, such as open cycle J-T coolers, split cycle systems where the compressor is fairly large and heavy to achieve greater reliability and ease in maintenance, liquefied air systems, etc. Therefore, because the mounting location and the overall design of the FLIR unit is still somewhat flexible, firm size and weight requirements were not applied during evaluation of the cryogenic refrigerator. Although large massive units were awarded fewer points than lighter units, no refrigerators were eliminated because of this parameter.

3.2.3 Mechanical Vibration of the Cryogenic Refrigerator Cold Head

In the past, refrigerator induced vibration has been a difficult problem for the infrared detector designers because the vibration impairs the resolution of the detector system. Because of the vibration problem, split cycle refrigerators such as J-T cycles have been preferred in detector applications. The Stirling cycle refrigerator was also used by condensing air or nitrogen at a remote location. The liquid was then transferred to the separated detector

package using a Leidenfrost transfer technique. From the discussions with FLIR manufacturers, the concern about mechanical vibration has practically disappeared. However, the movement of the cold head in relation to its mounting flange is still considered important. This movement is caused by pressure fluctuations in the cold head and by the natural frequency vibration of the cold finger. Radial movement of the head should be less than 0.013 mm at a frequency above 400 - 600 Hz. Axial movement is not as important because the depth of focus of the FLIR system is sufficient to maintain adequate resolution. Indications were that a refrigerator for the FLIR application is chosen for other qualities, and then the cold finger is redesigned, if necessary, to minimize movement.

4. Responses to Questionnaires

Table 1 is the compiled data received from manufacturers of cryogenic refrigerators. Data were requested for refrigerators having capacities of up to 10 watts within the temperature range of 15 to 100 K. Although the FLIR requirement was for a capacity of approximately one or two watts at 77 K, the additional information gained by requesting data for larger units could be used to evaluate a particular cycle or system. Reliability data was specifically requested, including verification of any supplied MTBF data and/or Mean Time To Repair (MTTR) data. Except in very few instances, reliability data was not verified and in most cases was not even supplied. The lack of data is understandable because maintenance data for many refrigerators running for long periods of time are required before good MTBF or MTTR data can be generated. In some systems, changes are still occurring, so reliability data from a predecessor is not completely applicable to the refined system. For the purposes of evaluation, MTBF values were assigned to each refrigerator based on limited life cycle tests, user experience with the system or similar systems, manufacturers' information, and theoretical predictions.

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators.

Identification Number	1	2	3	4
Refrigeration Temp. Range	40-120K	77K	50-120K	37-100K
Cycle	Modified Solvay	Joule Thomson (open)	Modified Solvay	Gifford McMahon
Working Fluid	Helium	Nitrogen	Helium	Helium
Lowest Temperature 1W/2W	70K/120K	77K/77K	77K/120K	38K/42K
High Pressure	1.79 MPa (17.7 Atm)	13.8-55.2 MN/m^2 (136-544 Atm)	2.59 MPa (25.5 Atm)	---
Low Pressure	0.45 MPa (4.42 Atm)	---	1.03 MPa (10.2 Atm)	---
Cooling Time No Load (min.)	5	0.15	5	4
Cooling Time Loaded (min.)	15	---	15	10
Temperature Stability	±0.05K (±0.09R) @ 6.7 Hz	+0.25K (+.45 R)	±0.05K (±0.09R) @ 7.5 Hz	---
Expander Rotational Speed	41.9 rad/s (400 rpm)	---	47.9 rad/s (450 rpm)	---
Compressor Rotational Speed	377.0 rad/s (3600 rpm)	---	377 rad/s (3600 rpm)	---
Compressor Lubrication	Oil Lubricated	---	Dry Lubricated	Oil Lubricated
Total Power Input	400 W	---	578 W	1000 W
Volts-Phase-Frequency	115V-1-60 Hz	---	115V-3-400 Hz	117V-3-60 Hz
Heat Rejected Near Detector	0	0	None	5 W Maximum
Cooling Medium	Forced Air	---	Forced Air	Forced Air
Ambient Temp. Requirements	4.4 to 43.3°C (40 to 110°F)	---	-40 to 57°C (-40 to 135°F)	4 to 43°C (40 to 110F)
Required Attitude	---	None	None	---
Compressor Volume	59,485 cm^3 (3630 in^3)	---	3248 cm^3 (198.2 in^3)	54,003.56 cm^3(3295.5 in^3)
Cryostat Volume	32 cm^3 (1.95 in^3)	1.07 cm^3 (0.065 in^3)	32 cm^3 (1.95 in^3)	1161.7 cm^3 (70.9 in^3)
Compressor Weight	27.2 kg (60 lbs)	---	5.76 kg (12.7 lbs)	38.5 kg (85 lbs)
Cryostat Weight	0.36 kg (0.8 lb)	0.072 kg (0.16 lb)	0.11 kg (0.25 lb)	4.98 kg (11 lbs)
MTBF	12,000 hours	Depends on Gas Quality	3651 hours	---
MTTR	---	6 hours	2.5-3.5 hours	1/2 hour
Maintenance Interval	4500 hours	---	1200 hours	6000 hours
Vibration	0.0005 cm PK-PK @ 7.5 Hz (0.0002 in)PK-PK @ 7.5 Hz	None	0.0005 cm PK-PK @ 7.5 Hz	---
Cost 50-100 units	\$1500-\$2000	\$400	\$4000	\$6000
Status	Production	Production	Production	Production

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	5	6	7	8
Refrigeration Temp. Range	11-22K	53-100K	30-80K	23-80K
Cycle	Gifford McMahon	Gifford McMahon	Gifford McMahon	Gifford McMahon
Working Fluid	Helium	Helium	Helium	Helium
Lowest Temperature 1W/2W	16K/22K	75K/100K	35K/42K	28K/33K
High Pressure	2.07 MPa (20.4 Atm)	1.45 MPa (14.3 Atm)	1.65 MPa (16.3 Atm)	2.07 MPa (20.4 Atm)
Low Pressure	0.62 MPa (6.12 Atm)	0.48 MPa (4.76 Atm)	0.48 MPa (4.76 Atm)	0.62 MPa (6.12 Atm)
Cooling Time No Load (min.)	300 to 11K in 60 m	25	15	10
Cooling Time Loaded (min.)	300 to 11K in 65 m	33	18	10
Temperature Stability	---	±0.1K (±0.2 R) @ 60 Hz	---	---
Expander Rotational Speed	15.1 rad/s (144 rpm)	15.1 rad/s (144 rpm)	15.1 rad/s (144 rpm)	15.1 rad/s (144 rpm)
Compressor Rotational Speed	377 rad/s (3600 rpm)	377 rad/s (3600 rpm)	377 rad/s (3600 rpm)	377 rad/s (3600 rpm)
Compressor Lubrication	Oil Lubricated	Oil Lubricated	Oil Lubricated	Oil Lubricated
Total Power Input	1200 W	450 W	830 W	1200 W
Volts - Phase - Frequency	120V -1-60 Hz	120V-1-60 Hz	120V-1-60 Hz	120V-1-60 Hz
Heat Rejected Near Detector	None	None	None	None
Cooling Medium	Forced Air	Forced Air	Forced Air	Forced Air
Ambient Temp. Requirements	-18 to 43°C (0-110°F)	-18 to 43°C (0 to 110° F)	-18 to 43°C (0 to 110°F)	-18 to 43°C (0 to 110° F)
Required Attitude	---	---	---	---
Compressor Volume	133,720 cm^3 (8160 in^3)	133,720 cm^3 (8160 in^3)	133,720 cm^3 (8160 in^3)	133,720 cm^3 (8160 in^3)
Cryostat Volume	1485 cm^3 (90.6 in^3)	1485 cm^3 (90.6 in^3)	1485 cm^3 (90.6 in^3)	1485 cm^3 (90.6 in^3)
Compressor Weight	79.4 kg (175 lbs)	14.97 kg (33 lbs)	15.8 kg (35 lbs)	79.4 kg (175 lbs)
Cryostat Weight	2.27 kg (5 lbs)	1.8 kg (4 lbs)	1.8 kg (4 lbs)	2.27 kg (5 lbs)
MTBF	---	---	---	---
MTTR	---	---	---	---
Maintenance Interval	3000 hours	3000 hours	3000 hours	3000 hours
Vibration	Perp. 2.5×10^{-5} cm @ 5 kHz Long. 2.5×10^{-6} cm @ 5 kHz	---	---	---
Cost 50-100 units	$4500	$3500	$3700	$4000
Status	Production	Production	Production	Production

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	9	10	11	12	13 *
Refrigeration Temp. Range	77K	50-80K	18-80K	78K	21-80K
Cycle	Joule Thomson	Gifford McMahon	Gifford McMahon	Stirling W. Leidenfrost	Stirling
Working Fluid	Nitrogen	Helium	Helium	Helium/Air	Helium
Lowest Temperature 1W/2W	77K/77K	66K/77K	26K/---	78K/78K	30K/---
High Pressure	13.8 MPa(136.1 Atm)	1.82 MPa (18 ATM)	1.82 MPa (18 Atm)	0.84 MPa (8.3 Atm)	0.96 MPa (9.5 Atm)
Low Pressure	0.10 MPa (1 Atm)	0.9 MPa (9 Atm)	0.91 MPa (9 Atm)	---	---
Cooling Time No Load (min.)	1.5	4	<30	<5	<10
Cooling Time Loaded (min.)	50	10 with 40 gms copper	<30	---	---
Temperature Stability	±0.1K (±.2 R)	---	±0.1K(±.2 R)@ 2.6 Hz	---	None Measured
Expander Rotational Speed	---	19.7 rad/s (188 rpm)	16.23 rad/s (155 rpm)	---	230.4 rad/s (2200 rpm)*
Compressor Rotational Speed	523.6 rad/s (5000 rpm)	1256.6 rad/s(12,000 rpm)	---	230.4 rad/s (2200 rpm)	230.4 rad/s (2200 rpm)*
Compressor Lubrication	Oil Lubricated	Oil Lubricated	Oil Lubricated	Non-Organic Grease	Non-Organic Grease
Total Power Input	400 W	525 W	800 W	660 W	530 W
Volts-Phase-Frequency	--- 400 Hz	200V-3-400 Hz	208V-3-400 Hz	115V-3-400 Hz	115V-3-400 Hz
Heat Rejected Near Detector	None	5 W Maximum	None	None	None
Cooling Medium	Forced Air	Forced Air	Forced Air	Forced Air	Forced Air
Ambient Temp. Requirements	---	-54 to 55° C (-65 to 131° F)	-40 to 55° C (-40 to 131° F)	-54 to 54° C (-65 to 130° F)	-54 to 54° C (-65 to 130° F)
Required Attitude	---	(±45° from Horizontal)	(±45° from Horizontal)	---	---
Compressor Volume	36,800 cm^3 (2246 in^3)	---	---	7876 cm^3 (480.7 in^3)	6496.5 cm3 (396.4 in^3)*
Cryostat Volume	24.6 cm^3 (1.5 in^3)	1475 cm^3 (90 in^3)	---	2.5 cm^3 (.153 in^3)	---
Compressor Weight	10.4 kg (23 lbs)	5.9 kg (13 lbs)	5 or 11.8 kg (11 or 26 lbs)	9.52 kg (21 lbs)	7.26 kg (16 lbs)*
Cryostat Weight	0.057 kg (0.125 lb)	0.91 kg (2 lbs)	2.04 kg (4.5 lbs)	0.045 kg (0.1 lb)	---
MTBF	6288 hours	---	---	---	---
MTTR	---	---	---	---	---
Maintenance Interval	2000 hours	2000 hours	1000 hours	500 hours	350 hours
Vibration	---	0.000025 cm PK-PK (0.00001 in) PK-PK	---	None at Detector	---
Cost 50-100 units	$9000	$8000	$9000	---	---
Status	Production	Production	Production	Production	Production

* Indicates an Integral Unit

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	14	15	16	17*	18
Refrigeration Temp. Range	78K	78K	23-80K	30-80K	40-80K
Cycle	Stirling W Leidenfrost	Stirling W Leidenfrost	Solvay Split	Stirling	Proprietary Split Cycle
Working Fluid	Helium/Air	Helium/Air	Helium	Helium	Helium
Lowest Temperature 1W/2W	78K/78K	78K/78K	30K/40K	41K/55K	48K/60K
High Pressure	0.841 MPa (8.3 Atm)	0.841 MPa (8.3 Atm)	1.82 MPa (18 Atm)	1.0 MPa (10 Atm)	2.03 MPa (20 Atm)
Low Pressure	---	---	0.71 MPa (7 Atm)	---	0.61 MPa (6 Atm)
Cooling Time No Load (min.)	< 5	< 5	5	4	7
Cooling Time Loaded (min.)	---	---	10	6	10
Temperature Stability	---	±0K	±0.01K (±0.02 R) @ 10 Hz	---	---
Expander Rotational Speed	---	NA	31.4 rad/s (300 rpm)	157.1 rad/s (1500 rpm)*	18 rad/s (172 rpm)
Compressor Rotational Speed	230.4 rad/s (2200 rpm)	230.4 rad/s (2200 rpm)	314.2 rad/s (3000 rpm)	157.1 rad/s (1500 rpm)*	377 rad/s (3600 rpm)
Compressor Lubrication	Non-Organic Grease	Non-Organic Grease	Dry Lubricated	Dry Lubricated	Dry Lubricated
Total Power Input	530 W	630 W	350 W	190 W	500 W
Volts-Phase-Frequency	115V-3-400 Hz	115V-3-400 Hz	110V-3-400 Hz	115V-3-400 Hz	115V-3-400 Hz
Heat Rejected Near Detector	None	0	0	190 W	10 W
Cooling Medium	Forced Liquid	Forced Air	Forced Air	Forced Air	Forced Air
Ambient Temp. Requirements	-54 to 71°C (-65 to 160°F)	-54 to 54°C (-65 to 130°F)	-31.6 to 82 °C (-25 to 180°F)	-54 to 55°C (-65 to 131°F)	-54 to 55°C (-65 to 131°F)
Required Attitude	---	---	None	None	None
Compressor Volume	3278 cm^3 (200 in^3)	7030 cm^3 (429 in^3)	7079 cm^3 (432 in^3)	1274 cm^3 (77.75 in^3)*	8685 cm^3 (530 in^3)
Cryostat Volume	2.5 cm^3 (0.15 in^3)	2.5 cm^3 (0.15 in^3)	2359 cm^3 (144 in^3)	NA	331 cm^3 (20.2 in^3)
Compressor Weight	5.4 kg (12 lbs)	7.94 kg (17.5 lbs)	4.1 kg (9 lbs)	3.2 kg (7 lbs)*	7.26 kg (16 lbs)
Cryostat Weight	0.045 kg (0.1 lb)	0.045 kg (0.1 lb)	3.4 kg (7.5 lbs)	---	1.36 kg (3 lbs)
MTBF	---	---	3000 hours (Projected	500 hours	1000 hours
MTTR	---	---	8 hours	10 hours	4.5 hours
Maintenance Interval	500 hours	500 hours	1500 hours	500 hours	1000 hours
Vibration	None at Detector	None at Detector	<0.001 in. all Directions (Estimated)	High	---
Cost 50-100 units	---	---	$3500[1]	---	$7800
Status	Production	Production	Production	Production	Production

* Indicates an Integral Unit

[1] Estimated cost after an initial investment to initiate production.

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	19*	20	21	22*	23*
Refrigeration Temp. Range	55-95K	50-250K	12-20K	60-80K	35-50K
Cycle	Stirling	Split Stirling	Solvay Process	Stirling	Stirling
Working Fluid	Helium	Helium	Helium	Helium	Helium
Lowest Temperature 1W/2W	83K/95K	110K/250K	18K/---	---	50K/---
High Pressure	0.61 MPa (6 Atm)	2.58 MPa (25.5 Atm)	0.304 MPa (3 Atm)	0.608 MPa (6 Atm)	1.01 MPa (10 Atm)
Low Pressure	NA	1.21 MPa (11.9 Atm)	0.101 MPa (1 Atm)	0.304 MPa (3 Atm)	0.507 MPa (5 Atm)
Cooling Time No Load (min.)	10	3	10	<10	<10
Cooling Time Loaded (min.)	15	8	15	NA	<10
Temperature Stability	---	±0.05K (±0.09R) @ 25 Hz	±0.15K (±0.3R) @ 5-8 Hz	±0.1K (±0.2 R)	±0.1K (±0.2 R)
Expander Rotational Speed	---	157.1 rad/s (1500 rpm)	42 rad/s (400 rpm)	188.5 rad/s (1800 rpm)*	188.5 rad/s (1800 rpm)*
Compressor Rotational Speed	138.23 rad/s (1320 rpm)*	157.1 rad/s (1500 rpm)	---	188.5 rad/s (1800 rpm)*	188.5 rad/s (1800 rpm)*
Compressor Lubrication	---	Dry Lubricated	---	---	Dry Lubricated
Total Power Input	60 W	80 W	460 W	90 W	200 W
Volts-Phase-Frequency	28V DC	24V DC	--- 60 Hz	208V-3-400 Hz	208V-3-400 Hz
Heat Rejected Near Detector	60 W	10 W	None	90 W	200 W
Cooling Medium	Forced Air	Forced Air	Forced Air	Forced Air	Forced Air
Ambient Temp. Requirements	-54 to 55° C (-65 to 131° F)	-40 to 73.8°C (-40 to 165°F)	---	-53.8 to 55°C (-65 to 131°F)	-53 to 55°C (-65 to 131°F)
Required Attitude	---	None	None	None	None
Compressor Volume	---	1793 cm^3 (109.4 in^3)	22,940 cm^3 (1400 in^3)	514.8 cm^3 (31.4 in^3)*	1274 cm^3 (77.7 in^3)*
Cryostat Volume	---	18 cm^3 (1.1 in^3)	521 cm^3 (31.8 in^3)	NA	NA
Compressor Weight	1.8 kg (4 lbs)*	2.5 kg (5.516 lbs)	7.25 kg (16 lbs)	<0.91 kg (2 lbs)*	< 2.27 kg (5 lbs)*
Cryostat Weight	---	0.098 kg (0.216 lbs)	1.8 kg (4 lbs)	NA	---
MTBF	500 hours	4000 hours (Projected)	500 hours	---	---
MTTR	5 hours	6 hours	---	---	---
Maintenance Interval	500 hours	1000 hours	---	500 hours	500 hours
Vibrations	---	---	---	---	---
Cost 50-100 units	$3000	$2000	$10,000	$4000	$6000
Status	Production	Under Development	Under Development	Under Development	Under Development

* Indicates an Integral Unit.

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	24*	25*	26*	27
Refrigeration Temp. Range	44-79K	51-86K	77-85K	77K
Cycle	Vuilleumier	Vuilleumier	Vuilleumier	Modified Stirling
Working Fluid	Helium	Helium	Helium	Helium
Lowest Temperature 1W/2W	55K/66K	63K/74K	85K/---	---
High Pressure	4.36 MPa (43 Atm)	4.36 MPa (43 Atm)	4.26 MPa (42 Atm)	2.84 MPa (28 Atm)
Low Pressure	NA	NA	NA	---
Cooling Time No Load (min.)	6.0	8.0	5.0	0.75
Cooling Time Loaded (min.)	---	---	---	---
Temperature Stability	None Measured	None Measured	None Measured	None Measured
Expander Rotational Speed	---*	62.83 rad/s (600 rpm)*	68.1 rad/s (650 rpm)*	138 rad/s (1320 rpm)
Compressor Rotational Speed	---*	62.83 rad/s (600 rpm)*	68.1 rad/s (650 rpm)*	138 rad/s (1320 rpm)
Compressor Lubrication	Dry Lubricated	Dry Lubricated	Dry Lubricated	Dry Lubricated
Total Power Input	350 W	340 W	120 W	80 W
Volts-Phase-Frequency	115V-3-400 Hz+250 W of 28VDC	115V-3-400 Hz +250 W of 28V DC	24V DC	28V DC
Heat Rejected Near Detector	350 W	340 W	120 W	0
Cooling Medium	Forced Air	Forced Air	Forced Air	Conduction to Structure
Ambient Temp. Requirements	-54 to 55.5°C (-65 to 132°F)	-54 to 54.4°C (-65 to 130°F)	-54 to 48.8°C (-65 to 120°F)	-54 to 71.1°C (-65 to 160°F)
Required Attitude	None	None	None	None
Compressor Volume	5962 cm^3 (363.8 in^3)*	7623 cm^3 (465.2 in^3)*	3323 cm^3 (202.8 in^3)*	403.6 cm^3 (24.6 in^3)
Cryostat Volume	NA	NA	NA	0.65 cm^3 (0.04 in^3)
Compressor Weight	3.63 kg (8 lbs)*	3.81 kg (8.4 lbs)*	2.63 kg (5.8 lbs)*	1.72 kg (3.8 lbs)
Cryostat Weight	NA	NA	NA	0.045 kg (0.1 lb)
MTBF	2500 hours	2500 hours	---	---
MTTR	---	---	---	---
Maintenance Interval	2500 hours	2500 hours	2000 hours	500-1000 hours
Vibration	---	---	---	---
Cost 50-100 units	---	---	---	---
Status	Operational	Operational	Operational	Under Development

* Indicates an Integral Unit

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	28	29*	30*	31*	32*
Refrigeration Temp. Range	25-77K	15-62K	50-77K	50-77K	50-77K
Cycle	Modified Stirling	Vuilleumier (Two Stage)	Vuilleumier	Vuilleumier	Vuilleumier
Working Fluid	Helium	Helium	Helium	Helium	Helium
Lowest Temperature 1 W/2 W	25K/77K[1]	23K/62K[1]	---	77K/77K	---
High Pressure	4.91 MPa (48.5 Atm)[2]	4.56 MPa (45 Atm)[2]	5.6 MPa (55 Atm)	7.09 MPa (70 Atm)	5.57 MPa (55 Atm)
Low Pressure	2.17 MPa (21.4 Atm)	NA	3.75 MPa (37 Atm)	3.75 MPa (37 Atm)	3.7 MPa (37 Atm)
Cooling Time No Load (min.)	1.5	6.5	12	10	12
Cooling Time Loaded (min.)	---	---	60	25	60
Temperature Stability	None Measured	None Measured	±0.01K (±0.02 R) @ 15 Hz	±0.01 K (±0.02 R) @ 15 Hz	±0.01K (±0.02 R) @ 15 Hz
Expander Rotational Speed	106.8 rad/s (1020 rpm)	62.8 rad/s (600 rpm)*	47.1 rad/s (450 rpm)*	47.1 rad/s (450 rpm)*	47.1 rad/s (450 rpm)*
Compressor Rotational Speed	106.8 rad/s (1020 rpm)	62.8 rad/s (600 rpm)*	47.1 rad/s (450 rpm)*	47.1 rad/s (450 rpm)*	47.1 rad/s (450 rpm)*
Compressor Lubrication	Dry Lubricated	Dry Lubricated	Dry Lubricated	Dry Lubricated	Dry Lubricated
Total Power Input	935 W	750 W	63 W	45 W @ 1 W of ref.	75 W
Volts-Phase-Frequency	115V-3-400 Hz	115V-3-400 Hz	28V DC	28 V DC	110V-1-400 Hz
Heat Rejected Near Detector	35	0	63	45 W	75 W
Cooling Medium	Forced Air	Forced Liquid	Forced Air	Forced Air	Forced Air
Ambient Temp. Requirements	-54 to 71.1°C (-65 to 160°F)	-54 to 71.1°C (-65 to 160°F)	-55 to 71°C (-67 to 160°F)	-55 to 71°C (-67 to 160° F)	-55 to 71°C (-67 to 160° F)
Required Attitude	None	None	None	None	None
Compressor Volume	16,793 cm^3 (1024.8 in^3)	17,830 cm^3 (1088 in^3)*	7964 cm^3 (486 in^3)*	3736 cm^3 (228 in^3)*	2595.7 cm^3 (158.4 in^3)*
Cryostat Volume	37.5 cm^3 (2.3 in^3)	---	---	---	---
Compressor Weight	13.2 kg (29 lbs)	8.845 kg (19.5 lbs)*	2.27 kg (5 lbs)*	1.36 kg (3 lbs)*	2.27 kg (5 lbs)*
Cryostat Weight	0.45 kg (1 lb)	---	---	---	---
MTBF	---	2500 hours	2000 hours (Projected	2000 hours (Estimate)	2000 hours (Estimate)
MTTR	---	---	8 hours (est.)	8 hours (Estimate)	8 hours (Estimate)
Maintenance Interval	2000 hours	2500 hours	1000 hours	1000 hours	1000 hours
Vibrations	---	---	---	---	---
Cost 50-100 units	---	---	$3500[3]	$2500[3]	$4500[3]
Status	Under Development	Operational	---	Under Development	Under Development

* Indicates an Integral Unit.

1 Two stage unit, refrigeration available at both temperatures simultaneously.

2 Fill pressure when refrigerator is warm.

3 Estimated cost after an initial investment to initiate production.

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	33*	34*	35*	36*
Refrigeration Temp. Range	45-77K	45-77K	45-77K	11.5-75K
Cycle	Stirling	Stirling	Stirling	Vuilleumier
Working Fluid	Helium	Helium	Helium	Helium
Lowest Temperature 1W/2W	---	77K/---	77K/---	<20K/<25K
High Pressure	0.76 MPa (7.5 Atm)	0.91 MPa (9 Atm)	0.91 MPa (9 Atm)	1.52 MPa (15 Atm)
Low Pressure	0.35 MPa (3.5 Atm)	0.51 MPa (5 Atm)	0.51 MPa (5 Atm)	0.91 MPa (9 Atm)
Cooling Time No Load (min.)	2.5	3	3	30[1]
Cooling Time Loaded (min.)	45	15	15	---
Temperature Stability	±0.0001K (±0.0002 R) @10 Hz	±0.0003K (±0.0005 R) @25 Hz	±0.0003K (±0.0005 R) @25 Hz	---
Expander Rotational Speed	62.8 rad/s (600 rpm)*	157 rad/s (1500 rpm)*	157 rad/s (1500 rpm)*	44 rad/s (420 rpm)*
Compressor Rotational Speed	62.8 rad/s (600 rpm)*	157 rad/s (1500 rpm)*	157 rad/s (1500 rpm)*	44 rad/s (420 rpm)*
Compressor Lubrication	Oil Lubricated	Dry Lubricated	Oil Lubricated	Oil Lubricated
Total Power Input	20 W	60 W	42 W	2700 W
Volts-Phase-Frequency	28V DC	DC	AC	100V DC and 28V DC
Heat Rejected Near Detector	20 W	60 W	42 W	2700 W
Cooling Medium	Solid Conduction	Air Natural	Air Natural	Water
Ambient Temp. Requirements	-34.4 to 65.5°C (-30 to 150°F)	-34.4 to 65.5°C (-30 to 150°F)	-34.4 to 65.5°C (-30 to 150°F)	0 to 50°C (32 to 122°F)
Required Attitude	None	None	None	None
Compressor Volume	2785.8 cm^3 (170 in^3)*	4670 cm^3 (285 in^3)*	4670 cm^3 (285 in^3)*	294,967 cm^3 (18,000 in^3)*
Cryostat Volume	---	---	---	---
Compressor Weight	3.63 kg (8 lbs)*	3.63 kg (8 lbs)*	3.36 kg (8 lbs)*	90.72 kg (200 lbs)*
Cryostat Weight	---	---	---	---
MTBF	10,000 hours (Estimate)	1000 hours	10,000 hours (Estimate)	10,000 hours (Estimate)
MTTR	8 hours (Estimate)	---	---	---
Maintenance Interval	8000 hours	500 hours	8000 hours (Estimate)	---
Vibrations	Long. 0.0005 cm DA @10 Hz Perp. 0.00012 cm DA @ 10 Hz	Long. <0.0005 cm DA @ 25 Hz Perp. <0.00012 cm DA @ 25 Hz	Long. <0.0005 cm DA @ 25 Hz Perp. <0.00013 cm DA @ 25 Hz	Long. 0.0005 cm DA @ 7 Hz None other axis
Cost 50-100 units	---	$4500	$5000	---
Status	Under Development	Testing Prototype	Proposed	Under Development

* Indicates an Integral Unit.

1 With hot side warmed-up; warm-up requires 60 min.

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	37*	38	39	40*
Refrigeration Temp. Range	50-77K	3.6K	80K	65K
Cycle	Vuilleumier	Claude Bypass	Brayton	Vuilleumier
Working Fluid	Helium	Helium	Nitrogen	Helium
Lowest Temperature 1W/2W	1.7 W @ 77 K	3.6K/---	---/80 K	---
High Pressure	3.04 MPa (30 Atm)	---	0.2 MPa (2.0 Atm)	6.89 MPa (68 Atm)
Low Pressure	1.72 MPa (17 Atm)	0.41 MPa (4 Atm)	---	6.1 MPa (60 Atm)
Cooling Time No Load (min.)	18	Several Hours	---	30
Cooling Time Loaded (min.)	21	---	240	---
Temperature Stability	---	±0.1K(±0.2 R)	±1.0K (±1.8 R)	---
Expander Rotational Speed	62.8 rad/s (600 rpm)*	20,943 rad/s (200,000 rpm)	10,996 rad/s (105,000 rpm)	41.9 rad/s (400 rpm)*
Compressor Rotational Speed	62.8 rad/s (600 rpm)*	8901 rad/s (85,000 rpm)	13,090 rad/s (125,000 rpm)	---
Compressor Lubrication	Dry Lubricated	Gas Bearing	Gas Bearing	Dry Lubricated
Total Power Input	175 W	8200 W	375 W	80 W
Volts-Phase-Frequency	24V DC	---	60 Hz	---
Heat Rejected Near Detector	175 W	0	0	80 W
Cooling Medium	Forced Air	---	Forced Water	Forced Water
Ambient Temp. Requirements	-54 to 52°C (-65 to 125 °F)	---	---	60°C (140°F)
Required Attitude	None	---	---	None
Compressor Volume	14,748 cm^3 (900 in^3)*	32,594 cm^3 (1989 in^3)	1180/cm^3 (71.8 in^3)	10,900 cm^3 (664 in^3)*
Cryostat Volume	---	73,800 cm^3 (4503 in^3)	12,960 cm^3 (79.1 in^3)	---
Compressor Weight	4.67 kg (10.3 lbs)*	27.2 kg (60 lbs)	2.13 kg (4.7 lbs)	7.48 kg (16.5 lbs)*
Cryostat Weight	---	19.5 kg (43 lbs)	4.6 kg (10.2 lbs)	---
MTBF	1000 hours	5000 hours (Estimate)	5000 hours	20,000 hours (Estimate)
MTTR	---	---	---	---
Maintenance Interval	1000 hours	---	---	---
Vibrations	Low	None Measured	None Measured	---
Cost 50-100 units	---	$80,000	$40,000	$35,000
Status	Under Development	Under Development	Under Development	Under Development

* Indicates Integral Unit.

Table 1. Performance and physical parameters of low capacity cryogenic refrigerators (Continued)

Identification Number	41 *	42
Refrigeration Range	75 K	77K
Cycle	Vuilleumier	Joule Thomson (Open)
Working Fluid	Helium	Nitrogen
Lowest Temperature 1W/2W	<75K/< 75K	77K/77K
High Pressure	6.89 MPa (68 Atm)	13.79-55.2 MPa (136-544 Atm)
Low Pressure	6.1 MPa (60 Atm)	0.1 MPa (1 Atm)
Cooling Time No Load (min.)	30	0.15
Cooling Time Loaded (min.)	60	---
Temperature Stability	± 1.0K (±1.8R)	---
Expander Rotational Speed	41.9 rad/s (400 rpm)*	---
Compressor Rotational Speed	---	---
Compressor Lubrication	Dry Lubricated	---
Total Power Input	280 W	---
Volts-Phase-Frequency	---	---
Heat Rejected Near Detector	280 W	0
Cooling Medium	Ammonia Heat Pipe	---
Ambient Temp. Requirements	---	---
Required Attitude	None	None
Compressor Volume	18,877 cm^3 (1152 in^3)*	---
Cryostat Volume	---	1.0 cm^3 (0.06 in^3)
Compressor Weight	9.1 kg (20 lbs)*	---
Cryostat Weight	---	0.072 kg (0.16 lb)
MTBF	20,000 hours (Estimate)	Depends on Gas Quality
MTTR	---	---
Maintenance Interval	---	---
Vibration	<0.0003 cm	None
Cost 50-100 units	$35,000	$400-$450
Status	Under Development	Production

* Indicates an Integral Unit.

Blanks in table 1 indicate that the requested data were not received, and MTBF numbers listed in the table were numbers supplied by the manufacturers. Each refrigerator has been assigned an identification number to be used throughout the report.

5. Evaluation of Refrigerators

Data received from manufacturers of cryogenic refrigerators were broken down into 21 separate categories, and points were assigned to each category, reflecting weighted-value as agreed upon by the NOL technical advisor and the author. In most cases, the evaluation was based on performance and physical size and weights of candidate refrigerators. However, in other instances, the judgment of the author was involved in selection of points for a particular category. Since the state of the art for small cryogenic refrigerators is changing rapidly, a definition of all categories, except those which were purely judgments of the author, will be made so that weighting factors and the total number of points may be changed to reflect changes in requirements and changes in performance, reliability, and availability of cryogenic refrigerators.

5.1 Mean Time Before Failure Values

Before defining the weighting factors associated with the categories, a discussion of the MTBF, the MTTR, and the manufacturers' suggested maintenance intervals is necessary. A total of 35 points, out of a possible 115 points, was assigned to these three categories because of the importance placed on reliability. It became apparent over the course of the study that actual values for these categories, except for the suggested maintenance interval, were either nonexistent or were based on experience with similar instead of identical systems and did not reflect actual field experience. The main reason for the lack of reliability data was a result of the limited number of refrigerators

in the field or because the refrigerators in the field were being used entirely differently than the projected FLIR use. Thus, the approach used to determine numbers for MTBF and MTTR was to define the numbers for a particular cycle or system based on theoretical predictions, interviews with manufacturers and users, and preliminary life test results. These MTBF numbers were then used throughout the evaluation where more firmly established numbers were not available. This approach, although not assigning directly measured reliability data to a particular refrigerator, did use numbers applicable to a fully developed refrigerator of the same type. In all instances, the manufacturers' suggested maintenance intervals were used in the evaluation.

Ten points were distributed to a particular refrigerator based on the suggested maintenance interval. Points were assigned based on the suggested interval for major maintenance such as compressor overhaul and not on simple maintenance such as helium recharging, or adsorber or filter change.

Twenty points were assigned to the MTBF evaluation. These points were based on the best available information and were assigned as follows:

1. Vuilleumier Refrigerators: The reliability data for VM machines is quite limited with very few machines in service. However, during interviews with people knowledgeable about VM machines, indications were that a 2500 hour MTBF was achievable, and test results seem to bear out this contention [9 and 10]. Thus, a 2500 hour MTBF was used for VM machines during the evaluation. The author's impression is, however, that the basic concept of the Vuilleumier cycle refrigerator should allow further increases in MTBF as the state of the art is advanced.

2. <u>Stirling Cycle Refrigerator:</u> Although the Stirling cycle refrigerator has many redeeming features, the reputation for reliability in low capacity units is not good. A MTBF of 500 hours was used during the evaluation of currently available low capacity Stirling cycle refrigerators. Several manufacturers of Stirling refrigerators feel that with some minor design changes the MTBF could be increased to 1000 - 2000 hours and with some major design changes such as rolling seals to allow oil lubrication of the crankcase while eliminating the contamination problem, would increase the MTBF to as high as 10,000 hours. To increase the MTBF of a Stirling cycle to this high value would make them appear quite attractive for the FLIR application.

3. <u>Gifford McMahon and Solvay:</u> The Gifford McMahon and Solvay cycles currently have the most operating hours under field conditions of any refrigerator type. The communication companies use this type of refrigerator for cooling parametric amplifiers and report a MTBF of greater than 6000 hours for a Gifford McMahon cycle using a slow turning, oil lubricated compressor. The cold head for this type of refrigerator is also large and has a slow moving displacer to obtain the extended life. The MTBF used for the larger machines incorporating large oil lubricated compressors was 6000 hours, while an MTBF of 4000 hours was used for smaller machines using higher speed, smaller oil lubricated compressors. For small units using dry lubricated or very high speed oil lubricated compressors, a 1500 hour MTBF was used for evaluation purposes.

5.2 Assignment of Weighting Factors

Table 2 shows the various categories, the number of points assigned to each category, and the method used to assign points during the evaluation. Refrigerators with one watt nominal capacity were evaluated identically with refrigerators having two watts nominal capacity, except for categories 4, 5, 6, 11 and 12 where the (a) subscripted categories were used to evaluate the one watt machines.

Table 2. Evaluation criteria for nominal 2 watt and 1 watt refrigerators

	Category	Total Points	Equation	Upper Limits	Lower Limits
1.	Refrigeration Temp. Range	4	None	4 pts if range 60K-75K 3 pts if range 60K-70K 2 pts if range 60K-65K 1 pt if single temperature	
2.	Cycle	None	None	Reject if open (for exception see Section 8)	
3.	Fluid	None	None	Reject if fluid is hazardous	
4.	Lowest temperature for a capacity of 2 W	4	Pts = 16 - 0.2 × (temperature corresponding to a 2 W capacity, K)	T >75K, 0 points T >80K, Reject	T <60K, 4 points
4a.	Lowest temperature for a capacity of 1 W	4	Pts = 16 - 0.2 × (temperature corresponding to a 1 W capacity, K)	T >75K, 0 points T >80K, Reject	T <60K, 4 points
5.	Cooldown time with load for a capacity of 2 W	4	Pts = 5 - 0.2 × (cooldown time with load, m)	Time >25 m, 0 points Time >30 m, Reject	Time < 5 m, 4 points
5a.	Cooldown time with a load of 1 W	4	Same as 5	Same as 5	Same as 5
6.	Total power required (nominal 2 W refrigerator)	4	Pts = 4.5 - 0.005 × (total power required in watts)	Power >900 W, 0 points Power >1500 W; Reject	Power <100 W, 4 points
6a.	Total power required (nominal 1 W refrigerator)	4	Pts = 5.143 - 0.01143 × (total power required in watts)	Power >450 W, 0 points Power >1500 W, Reject	Power <100 W, 4 points
7.	Heat rejected at detector	4	Pts = 5 - 0.01 × (heat rejected, W)	Heat >500 W, 0 points No reject criteria	Heat <100 W, 4 points
8.	Cooling medium	4	None	No external cooling source such as a coolant fluid, 4 pts	If an external coolant fluid is required, 0 points.
9.	Operating temperature requirements upper temperature limit	2	Pts = -2.909 + 0.091 × (higher operating temperature limit, °C)	Upper temperature >54°C, 2 points	Upper temperature <32°C, 0 points

Table 2. Evaluation criteria for nominal 2 watt and 1 watt refrigerators (continued)

	Category	Total Points	Equation	Upper Limits	Lower Limits
10.	Operating temperature limits lower temperature limit	2	Pts = -1.5079 - 0.10526 x (lower temperature limit, °C)	Lower temperature > 15 °C, 0 points	Lower temperature < -34 °C, 2 points
11.	Envelope size, cm^3 Integral unit (2 W nominal capacity)	5	Pts = 6 - 0.001 x (volume, cm^3)	Volume > 6000 cm^3, 0 points	Volume < 1000 cm^3, 5 pts
	Envelope size Split unit				
	Compressor size	2	None	Volume > 10,000 cm^3, 0 pts No reject criteria	Volume < 10,000 cm^3, 2 pts
	Cryostat size	3	Pts = 3.6 - 0.0006 x (volume, cm^3)	Volume > 6000 cm^3, 0 points	Volume < 1000 cm^3, 3 pts
11a.	Envelope size, cm^3 Integral unit (1 W nominal capacity)	5	Pts = 5.903 - 0.00257 x (volume, cm^3)	Volume > 2300 cm^3, 0 points No reject criteria	Volume < 350 cm^3, 5 points
	Envelope size Split unit				
	Compressor size	2	None	Volume > 10,000 cm^3, 0 pts	Volume < 10,000 cm^3, 2 pts
	Cryostat size	3	Pts = 3.542 - 0.00154 x (volume, of cold head, cm^3)	Volume > 2300 cm^3, 0 pts No reject criteria	Volume < 350 cm^3, 3 pts
12.	Envelope weight, kg (2 W nominal capacity) Integral unit	4	Pts = 8 - 1.333 × (wt, kg)	Wt, kg > 6, 0 points	Wt, kg < 3, 4 points
	Envelope weight, kg Split unit				
	Compressor weight	1	None	Wt, kg > 8, 0 points	Wt, kg < 8, 1 point
	Cryostat weight	3	Pts = 6 - (wt, kg)	Wt, kg > 6, 0 points	Wt, kg < 3, 3 points
12a.	Envelope weight, kg (1 W nominal capacity) Integral unit	4	Pts = 6.6667 - 1.333 × (wt, kg)	Wt, kg > 5, 0 points	Wt, kg < 2, 4 points
	Envelope weight, kg Split unit				
	Compressor weight	1	None	Wt, kg > 8, 0 points	Wt, kg < 8, 1 point
	Cryostat weight	3	Pts = 5 - (wt, kg)	Wt, kg > 5, 0 points	Wt, kg < 2, 3 points

Table 2. Evaluation criteria for nominal 2 watt and 1 watt refrigerators (continued)

	Category	Total Points	Equation	Upper Limits	Lower Limits
13.	MTBF, h	20	Pts = 13.133 × $\log_{10}$ (MTBF) - 32.532	MTBF >10,000 h, 20 points	MTBF <300 h, 0 points
14.	MTTR, h	5	None	Selected according to judgement	Selected according to judgement
15.	Maintenance Interval, (MI), h	10	Pts = 6.5665 × $\log_{10}$ (Maint. Int., h) - 16.266	MI >10,000 h, 10 points	MI, h <300, 0 points
16.	Cost	4	Pts = 5.71 - 0.000571 × (cost)	Cost <\$3000, 4 points	Cost >\$10,000, 0 points
17.	Vibration	5	None	Selected according to judgement	Selected according to judgement
18.	Manufacturer's credibility	10	None	Selected according to judgement	Selected according to judgement
19.	Interfacing characteristics	6	None	Selected according to judgement	Selected according to judgement
20.	EMI	4	None	Selected according to judgement	Selected according to judgement
21.	Development risk	10	None	Selected according to judgement	Selected according to judgement

Most of the categories are self-explanatory. The cooldown time used in category 4 is the cooldown time for an average thermal mass of 13 joules/K for a nominal two watt machine and one half that or 6.5 joules/K for a nominal one watt machine. The cooldown time for the machines cooling an average thermal mass of 13 joules/K was requested during the survey. Several companies did not supply the requested information, and cooldown time for these refrigerators was calculated by estimating a room temperature capacity for the refrigerator to be three times the capacity at 77 K. A linear decrease in capacity was then assumed between 300 and 70 K. Many inaccuracies are present in this method of evaluation, but the estimate was only necessary in a few instances, and the resulting numbers are probably conservative [11].

Categories 7, 11, and 12 are weighted against the integral unit. In category 7 (Heat Rejected at Detector), additional heat rejected near the detector was assumed to be a problem because of the electronics located within this closed area. In categories 11 and 12, the size and weight of the unit mounted in the detector enclosure was considered to be more of a problem than transversing the two gimbals with flexible connections. Because of these assumptions, split units did acquire a higher rating than an integral unit located entirely within the detector enclosure.

In categories 17, 18, 19, 20, and 21, points were assigned according to the author's best judgment. In category 17 (Vibration), judgment was necessary because very little information was supplied by the manufacturers. Points were assigned according to the class of machine. Stirling cycle machines without rhombic drive were rated lowest, while Joule Thomson cryostats were assigned the most points, and other machines were scattered between the two extremes.

In category 18 (Manufacturers Credibility), the manufacturer was assigned 10 points, unless in the author's judgment, the projected performance for a refrigerator under development appeared to be overly optimistic.

In category 19 (Interfacing Characteristics), the apparent advantage of split refrigerators was removed because of the reduced number of points assigned to these units. Both the size of the cold head and the difficulty of crossing between either one gimbal or two gimbals were used to determine the number of points assigned to a particular refrigerator.

In category 20 (Electro-Magnetic-Interference), the points were assigned according to the need for power near the detector and the judged manufacturers' experience in shielding against EMI. Therefore, units that had been used or were specifically designed for use with an infrared detector rated highest.

Points in category 21 (Development Risk) were assigned based on the number of refrigerators in the field and the number of estimates of other performance parameters required during the evaluation. A relatively low cost per unit, particularly for the VM machine, was assigned during the evaluation. These costs were used because of estimates and in some instances guesses received during discussions with possible users and the manufacturers. Because of the price uncertainty, the number of points assigned in category 21 were reduced slightly for VM machines.

6. Results of the Refrigerator Evaluation

Tables 3 and 4 show the total number of points assigned during the evaluation to refrigerators acceptable for a nominal one watt heat load and a nominal two watt heat load. The refrigerators receiving

Table 3. Evaluation results for one watt refrigerators.

Refrigerator I. D. Number	No. of Points
1	88.09
10	84.82
4	84.81
3	84.74
9	82.92
18	81.84
7	80.25
35	79.91
16	75.20
6	75.12
19	74.86
17	71.59
31	71.51
24	71.00
25	69.96
15	68.90
12	67.90
23	65.85
14	64.90
37	61.20
34	60.12

Table 4. Evaluation results for two watt refrigerators.

Refrigerator I. D. Number	No. of Points
4	86.35
10	84.15
18	83.54
7	81.78
16	78.49
17	75.40
31	72.28
24	70.78
25	68.56
15	68.25
23	67.85
12	67.10
14	64.75
37	60.20

the highest number of points are considered best suited for FLIR device cooling applications. As could be predicted from the weighting factors, the refrigerators to which the highest MTBF values were assigned were rated highest even though the compressors, and occasionally the cold heads of these units, were heavy and physically large. The higher rating of these large units was the result of assigning a higher number of points to the MTBF and the maintenance interval category than the number of points deducted for interfacing problems associated with the large size, split components. For most installations aboard ship, the use of a split refrigerator utilizing a physically large compressor is acceptable; however, if the complications involved in using a split cycle are prohibitive, an integral unit receiving a high number of points could be utilized. Based on the current size of the prototype FLIR unit, even the large size compressors appear small and could be mounted in the superstructure of the ship (position B, figure 10).

7. Cryogenic Refrigerators Under Development for Space Applications

Several cryogenic refrigerators are currently being developed for space application. These refrigerators are being developed at considerable cost to give long term, unattended reliable operation. The projected cost of this type of unit is \$35,000 to \$40,000 per unit after development costs, and they have projected unattended life times of one to five years. The units include several Vuilleumier units, several turbo machinery Brayton cycle units, and a rotary reciprocating Brayton cycle unit. Even though the projected long term reliability of these units would make their application in a FLIR unit desirable, the machines were not included in the previous evaluation because of their low level of development, the high cost of the machines, and their unproven long term reliability.

The VM refrigerators specifically designed for space applications incorporate closer tolerances and more expensive materials for the seals,

rider rings, and bearings. One design incorporates rolling seals with an oil lubricated crankcase to obtain extended life. Another design uses noncontacting dynamic seals (not subject to wear) together with inorganic bearing materials, providing minimum contamination of the working fluid to obtain extended life. Another design is very similar to the refrigerators designed for shorter life except that more expensive seal and rider ring materials are used. Refrigerators identified by numbers 40 and 41 in table 1 are examples of VM machines designed for space applications. More technical descriptions of the VM machines developed for space applications can be obtained elsewhere [12, 13, 14].

Reversed Brayton cycle refrigerators using turbomachinery are under development for space applications; they incorporate gas bearing supported rotary compressors and expansion turbines. The complexity of gas bearing systems for use in low capacity refrigerators is apparent because even though development of this type of unit has been funded since the 1960's, no low capacity machines of this type are in use in a field application. Gas bearings were chosen for use in the Brayton cycle because they allow the high rotational speeds necessary for high efficiency in the turbomachinery, they have long life because there is no physical contact in the bearings during operation, and they are contamination free because the working fluid is used as the bearing lubricant instead of a foreign material. The low capacity turbomachinery units have a relatively low efficiency at low temperatures (below 60 K); however, higher efficiency is possible at 77 K. The turbomachinery units have a very long cooldown time (2 to 3 hours is estimated), and if used in the FLIR application, they would need to run continuously to provide refrigeration within a reasonable time after a need is

established. Refrigerator numbers 38 and 39 are examples of two prototype machines constructed using turbocompressors and expanders. Because of the remaining problems with the turbomachinery and the availability of less complex refrigerators for use with the FLIR system, these units were not considered for use with the FLIR unit. Additional information [15, 16] on turbomachinery refrigerators is available.

The rotary-reciprocating refrigeration system under development [17] also uses gas lubricated bearings to support the compressor and expansion pistons. The compressor and expansion pistons are rotated to maintain the gas film in the supporting bearings and to provide the correct valving sequence during the cycle while a superimposed reciprocating motion provides the high and low pressure in the cycle. An overall thermal efficiency for the rotary reciprocating system should be higher than a comparable turbomachine system. Because of the design parameters of nonattended operation for long periods of time for this refrigerator, the complex gas bearing design, and the complex electrical control system required for operation, the cost of a rotary reciprocating refrigerator would be high, and the system was not considered for the FLIR application.

8. The Open Cycle Joule-Thomson Refrigerator

During the course of the study, it was discovered that ships for which FLIR units are planned already have an onboard high pressure air supply. The pressure available is between 20 and 30 MPa making the use of this supply for an open cycle Joule-Thomson refrigerator possible. Redundant, and in most cases, nonlubricated compressors are aboard ships providing the reliable high pressure supply.

Calculated air flow through the Joule-Thomson refrigerator for a heat load of one watt is approximately 0.00315 gm/sec and twice that or .00631 gm/sec for a two watt heat load for a 26 MPa supply. Purification systems are available to remove impurities from an air stream to allow the operation of the refrigerator.

The suggested approach would be to use a large purifier connected to the high pressure supply and then flexible connections to the gimbaled FLIR unit. A tube with an inner diameter of 1.5 mm could be used for most of the connecting piping. Flexible tubing of suitable size is available to use through the gimbals. A small purifier located adjacent to the cold head would remove all remaining impurities before the air enters the cryostat. The Joule-Thomson cryostat should be a demand flow type (numbers 2, 42 and 43), which allows quick cooldown and yet conserves the air supply during steady state operation. Figure 11 is a schematic of the suggested system.

If the open cycle Joule-Thomson refrigerator had been evaluated along with the closed cycle refrigerators, it would rate near the top for both the one watt and two watt refrigerators because of the high reliability of the high pressure air supply.

Projected functional qualities of the Joule-Thomson refrigerator are: (1) the lowest temperature for a two watt or one watt load would be 78 K; (2) cooldown time with load using a demand flow Joule-Thomson valve would be approximately four minutes; (3) no heat would be rejected near the detectors--in fact, a low flow of ambient temperature reject air would be available for additional cooling within the gimbaled enclosure; (4) operating ambient temperature limits would not be important except to the supply compressors; and (5) interfacing problems are not severe except for the flexible connections leading through the gimbals.

The major problem with an open Joule-Thomson refrigerator resides in the purification system. The suggested maximum impurities in the purified air for successful operation of a Joule-Thomson refrigerator are as follows:

Water Vapor - Dewpoint: -33 C (-92F) or lower

Carbon Dioxide: 1 part per million by weight or less

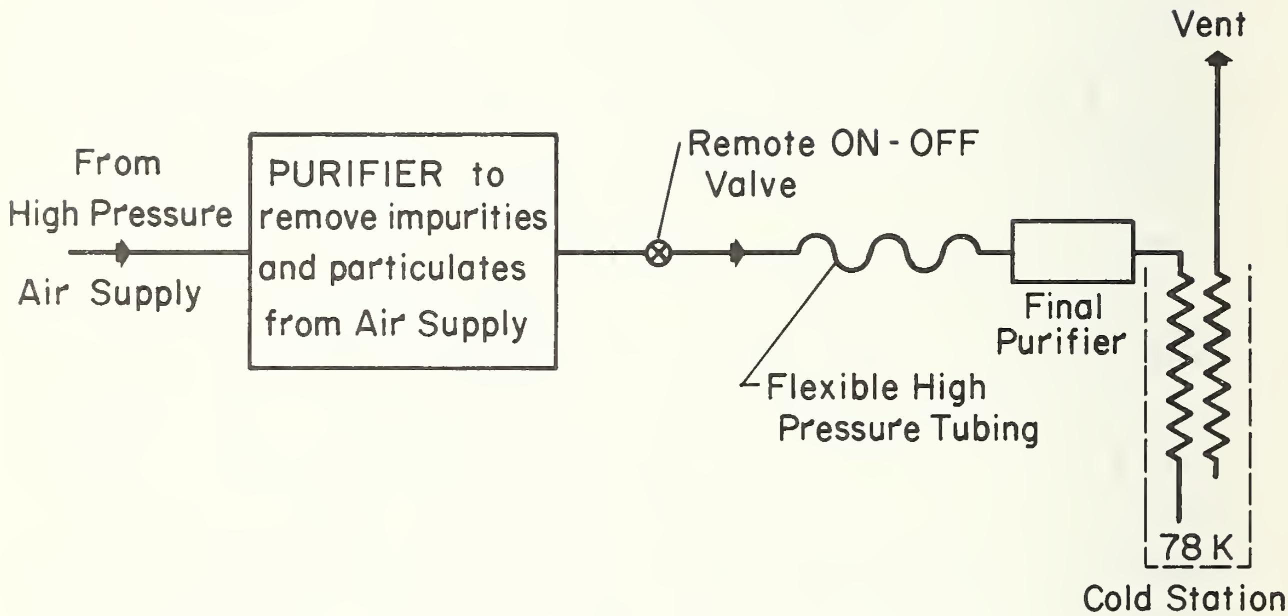


Figure 11. Shipboard, open cycle, Joule-Thomson refrigerator.

Hydrocarbons: 1 part per million by weight or less

Solid Particles: 5 microns maximum size.

Several manufacturers produce applicable high pressure purification systems capable of achieving the required purity. A typical system is available for approximately $4000 that is reportedly adequate for purifying 25.5 kg of air. At the calculated flow, this system would last for 2248 hours of steady state operation at a one watt nominal heat load. After depletion of the chemical desiccants, they must be replaced at a cost of $8.36 per charge. Similar purification systems have been proposed for use with oxygen-nitrogen generators and there is no reason to believe the system could not be acceptable for use on board ship.

9. Experience of Other Users of Cryogenic Refrigerators

Discussions were held during visits to Wright Patterson Air Force Base and the Night Vision Laboratories with Air Force and Army personnel working with small cryogenic refrigerators. The discussions at both facilities were largely directed towards small refrigerators with application to IR detectors. These refrigerators were being developed either under contract to outside manufacturers or in-house at the facilities. A large amount of money has been spent in the last several years to develop small, low weight, and reliable refrigerators; personnel at both facilities had high expectations that reliable small cryogenic refrigerators would soon be available at acceptable prices.

The Air Force believes that sufficient data has currently been collected to indicate an MTBF for the VM refrigerator of at least 2000 hours. The only maintenance required during the operating period is the addition of helium every 500 - 1000 hours. The major difficulty with reaching the 2000 hour MTBF involved the electronic heater controls

and power inverter for the drive motor. They felt these problems have now been solved, allowing the achievement of the 2000 hour MTBF. They also felt that the life of the machine can be extended beyond 2000 hours as experience is gained with the hot rider rings, one of the remaining major problems. The Air Force also felt the VM machines would approach a cost of $3000 per unit after an initial investment to set up an assembly line to produce large quantities of the machines. They are trying to standarize on a particular refrigerator configuration that could be used throughout at least the Air Force, allowing the machine to be mass produced in large quantities. The proposed configuration would be a split cycle VM; i. e. , the free piston expander would be connected to a thermal compressor by a single length of tubing.

The Air Force felt that the vibration and temperature stability of the VM was compatible with infrared detectors. Interfacing was also no problem, especially with the split cycle VM. One problem does exist with the VM, that is the refrigeration capacity of the unit decreases during its lifetime. This degradation in performance is not unique with the VM, as other types of refrigerators also display a degradation in performance after long term use. Insufficient data exists at this time to define the degradation in terms of the time of operation; however, purchasing a refrigerator with approximately 20% extra capacity when new allows sufficient remaining capacity at the end of its life.

Some refrigerators currently used for airborne application by the Air Force were showing an MTBF of less than 200 hours, while others were showing an MTBF of approximately 500 hours. These relatively low MTBF figures when compared to long life obtained in laboratory tests for the same refrigerators were attributed to the extreme environmental conditions aboard the aircraft.

The Army was not as firm on which refrigerator they would finally use. They felt the application should determine the choice of refrigerator. In some instances, where a heat source was available, the VM machine was the logical choice, while if efficiency and extremely small size were requirements, the Stirling cycle refrigerator may be more suitable. They also felt that if the required performance was specified, reputable manufacturers in the field of cryogenic refrigerators could produce an acceptable refrigerator. They thought the manufacturer could produce a machine to match a required MTBF figure; however, one must be willing to pay for long MTBF requirements. The Army is involved with development work on an integral Stirling cycle refrigerator utilizing rhombic drive to minimize vibrations, a split cycle Stirling cycle refrigerator utilizing a free piston expander, and a low cost VM machine.

Telephone contact was established with the Naval Electronics Laboratory who is in the process of purchasing 14 VM machines to be mounted with detectors in the superstructure of the ship. They chose VM machines because they will be installed in areas not accessible for repair, and they felt the VM was the most reliable refrigerator. They felt the VM machine was capable of long term operation without maintenance, though the machines being purchased were to have a disappointing guaranteed 3000 hour MTBF. The detector unit was designed to rotate continuously in a single direction, so beside rider rings for electrical interconnections, the additional complication of interconnecting a large split cycle refrigerator was not considered practical. The continuous rotation was one reason for not using an open cycle Joule-Thomson air refrigerator; the other reason was, in the opinion of the Navy, the high pressure air could not be purified sufficiently

to allow for continuous reliable operation of the Joule-Thomson refrigerator. The judged deficiency of the purification process was based on experience with airborne Joule-Thomson refrigerators where the cryostats would begin to plug after 6 hours of continuous operation, requiring a warm-up to clear the Joule-Thomson throttling valve of contaminates before the unit could again be cooled down.

Telephone contact was also established with a communication company using cryogenic refrigerators to cool parametric amplifiers on ground based communication antennas. They were extremely concerned about reliability; hence, two cryogenic refrigerators are used at each site so that a failure of either refrigerator would not close down the antenna system. The Gifford-McMahon refrigerators, currently installed, have slow running oil lubricated compressors. Their experience has been very good with a reported MTBF of 6500 hours for 24 refrigerators operating over the last 4 years.

They felt that this MTBF was actually low because several failures used to calculate the MTBF could be traced to vendor problems with associated equipment, and if these failures were eliminated from consideration, the calculated MTBF would increase to 9000 hours. They were not familiar with small size refrigerators normally used with infrared detectors, but said that if reliability were a requirement large oil lubricated compressors together with a Gifford McMahon type split cycle would achieve extended MTBF times.

10. Life Cycle Costs

To exactly predict the cost of operation for a cryogenic refrigerator used in the FLIR system is quite difficult. However, based on initial cost and cost of repair parts, at least the cost of repairs can

be determined based on the manufacturers' suggested maintenance intervals and the normal costs of the repairs. The maintenance intervals will, of course, differ with experience in the field, and the costs of repairs will also change as bulk purchases reduce the costs. In the following section, the costs of maintenance for three systems will be compared based on the recommended maintenance intervals supplied by the manufacturers. Some of the costs are of course estimates because very little experience has occurred with one of the types of refrigerator.

The three systems to be compared are a system with a large oil lubricated compressor and a Gifford-McMahon split cycle cold head, the open cycle Joule-Thomson refrigerator using high pressure air already available from a supply aboard ship, and a Vuilleumier refrigerator. The Vuilleumier refrigerator presented the most difficulty when estimates of repair costs were to be determined. The initial cost of Vuilleumier refrigerators is not known accurately, let alone the repair costs.

10.1 The Gifford-McMahon Refrigerator Costs

In order to make the cost estimates, repairs for refrigerator number 4 were used to represent this type of refrigerator, although the costs involved with any similar refrigerator that employs an oil lubricated air conditioning compressor converted to helium use and a separate cold head should be nearly the same. The initial cost for the refrigerator without any additional costs involved with interfacing the refrigerator is $6500. An adsorber used with the compressor should be changed every 3000 hours of operation at a cost of $95 from the factory. The cold head in the refrigerator can be refurbished in less than one hour, in place, for less than $100. The cold head refurbishment

will be accomplished every 6000 hours for this cost analysis, although refrigerators will probably run much longer without maintenance. The oil lubricated compressor has run as long as 10,000 - 20,000 hours without any maintenance; however, for our costing analysis, the compressor will be replaced every 8000 hours at a cost of $500. The basis used to calculate costs is half-time operation for five years or a total of 21,900 hours. Thus, the total cost for the operation of a single refrigerator for five years is:

$$6500 + \frac{21900}{3000}(95) + \frac{21900}{6000}(100) + \frac{21900}{8000}(500) = \$8927.25 .$$

10.2 The Open Cycle Joule-Thomson Refrigerator Costs

The primary cost incurred by the open cycle Joule-Thomson refrigerator is the cost of purifying the air. The initial costs used in the calculations were $4000 for the air purifier, $400 for the Joule-Thomson cryostat, and $2000 for the interconnecting plumbing, including a remote on-off high pressure valve. During the calculation, the purifier life was five years with the only cost associated with the purifier being that of replacing the chemicals. The Joule-Thomson cryostat will probably last much longer; however in the cost calculation, it will be replaced every 8000 hours. The on-off valve and connecting plumbing were replaced once during the five year life. The purifier materials must be replaced every 2200 hours for a one watt heat load at a cost of $8.36. Thus, the resulting cost of maintenance for a one watt refrigerator, not including any maintenance of the supply compressors, is:

$$2000 + 4000 + 400 + \frac{21900}{2200}(8.36) + \frac{21900}{8000}(400 + 100) = \$7851.97 .$$

The cost for a two watt refrigerator would be:

$$2000 + 4000 + 400 + \frac{21900}{1100}(8.36) + \frac{21900}{8000}(400 + 100) = 7935.19 .$$

10.3 Cost of Operating a Vuilleumier Refrigerator

The operating cost for a VM machine over the five year period is much harder to estimate. The initial cost of $3000 for a VM refrigerator used during the evaluation may have been an optimistic number, as a $3000 VM refrigerator is not yet available. A possible feature of a low cost VM may include a welded construction to insure helium leak tight sealing; this construction allows for easy initial assembly but does not allow for easy overhaul. In order to make a comparison of costs (based on information that a VM machine is well worn after 2500 hours of operation), the overhaul costs chosen for a VM were one-third to one-half its initial cost. This estimate makes the cost of maintenance high, but when compared to overhauling an internal combustion auto engine, the costs would appear to be reasonable. Costs for five years of operation if overhaul costs are one-third the initial cost are:

$$3000 + \frac{21900}{2500}(1000) = \$11,760.$$

Based on the estimate of the repair costs being one-half the initial purchase cost, the maintenance cost for a single machine becomes

$$3000 + \frac{21900}{2500}(1500) = \$16,140 .$$

11. Discussion

Because of the early developmental stage of a shipboard FLIR unit, volume, weight, and physical shape restrictions for the cryogenic refrigerator are not firmly established. Also, any type of refrigerator

is capable of meeting the functional requirements of the FLIR unit with minimal modification. Therefore, selection of a refrigerator should be made primarily on the basis of reliability and cost. Several small size and low weight refrigerators under development promise high reliability; however, other physically larger units have been used extensively and show extended MTBF's.

The recommendations for a cryogenic refrigerator will be divided into two groups. Group 1 will be for a split refrigerator with the compressors located at A or B in figure 10. Group 2 will be a recommendation for an integral refrigerator located at C in figure 10.

The first recommendation in group 1 is the Gifford McMahon cycle refrigerator with the large oil lubricated compressor which has been so successful for the communication companies. This refrigerator type is identified by numbers 1, 4, and 7 in table 1. Because the compressor for these refrigerators is large, it will be located off the gimbals at B in figure 10. Several problems remain in the use of oil lubricated compressors. One problem is the effect of roll and pitch of the ship on operation of the oil lubricated compressor. Manufacturers specifications indicate a maximum deviation from horizontal of ± 10 degrees for oil lubricated compressors; however, this specification is for a permanent installation and whether or not a temporary roll to 40 degrees would seriously effect compressor operation is debatable. Obviously, many oil lubricated compressors such as air compressors and air conditioning compressors are used aboard ship. The oil-lubricated compressors used with the refrigerator numbers 1, 4, and 7 are air conditioning compressors converted to helium use and probably can be used aboard ship without any modification. Manufacturers of compressors who were contacted during the study indicated that the roll and pitch problem was not that severe and most compressors

could be used either directly or with some simple modification. Another problem with using the large system involves the flexible connection between the remote compressor and the internal cold head. the proto-type FLIR system currently being purchased by the Navy has the compressor located on the outer gimbal with flexible connections to the inner gimbal. Because the rotation of the outer gimbal is currently limited to 360 degrees, a flexible connection between a compressor located at position B, figure 10 and the inner gimbal is possible. However, if continuous rotation of the outer gimbal is visualized, the use of a remote compressor is probably no longer acceptable due to the increased complexity of a nonleaking rotating joint.

The second recommendation in group 1 is the open cycle Joule-Thomson refrigerator. This refrigeration system was the least expensive unit considered for operation over five years, but besides the problem of the flexible connections required between the remote compressors and the internal gimbal, there is the added problem of purification of the air being used by the system. Although the technology of purification is highly developed, only actual experience of long term use for a similar system could prove the long term reliability of the system. Some users of Joule-Thomson systems expressed some doubt that continuous operation without plugging was within the state of the art while others felt the problem was readily solvable. Present use of the low capacity Joule-Thomson systems is for relatively short periods of time (up to six hours), and during these periods, some plugging of the Joule-Thomson valve has been evident with airborne Joule-Thomson refrigerators. The airborne systems, however, have limited space for the purifier and do not reflect the performance of a larger purification system of modern design. The only reason the Joule-Thomson refrigerator was placed second was because of the slightly unknown nature of the purification system.

The third recommendation for a refrigerator in group 1 would be to use a split cycle but with the compressor located on the outer gimbal. Three cycles fall under this recommendation and are identified by numbers 9, 10, and 18 in table 1. This type of refrigerator cannot exhibit the extended MTBF figures of the previous systems because of the state of the art of nonlubricated machinery and/or the high rotational speeds required in miniaturized oil lubricated equipment. The compressor orientation is not a problem with non-lubricated compressors, but the flexible connection between outer gimbal and the inner enclosure still exists. The flexible connection should not be a problem because the inner gimbal travel will be limited to a maximum travel of 90 degrees.

The last recommendation in group 1 is the split cycle VM refrigerator with the compressor located on the outer gimbal. The line length between the compressor and the cold head must be limited to less than 1 meter to maintain an acceptable performance. If a longer line is required by physical constraints, the split VM is not acceptable. The split VM is currently being developed so no current model appears in table 1.

Integral refrigerators in group 2 present a more difficult rating task. The use of the less reliable small integral refrigerator should be limited to situations where refrigerators from the first group are not feasible due to the complexity of flexible connections or other problems. At least based on the present rating system, an integral unit cannot be rated as highly as a split system because of rejected heat within the FLIR enclosure, the large physical size and weight of an integral unit when compared to the cold head size and weight of a split

cycle, and the low or unproven reliability (MTBF's) of many of the small integral units. Several small units do have respectable MTBF's; however, the maintenance interval, for instance, the compressor overhaul interval is relatively short (500 hours or less).

The highest or 1st rating to an integral unit would have to be for a Vuilleumier cycle refrigerator. Refrigerators numbered 24 and 31 are the best current models of the VM machine available for this application. The VM has many ardent supporters as evidenced by the large amounts of development money spent. The basic cycle and design allows for low seal wear and reduced contamination because the electric drive motor is not in the working volume. Also, the bearings have low loading, resulting in longer life when compared to a Stirling cycle refrigerator. The cost of a VM system is also a question at this time with some development being funded to produce a machine costing less than $3000 with a MTBF design goal of 3000 hours. The quoted cost per machine is not the total cost of the machine because a significant investment, not included in the $3000 cost, is required to set up a production line for the developed refrigerator. The manufacturers' suggested maintenance intervals for these machines are also approximately 3000 hours.

The second choice in group 2 is the Stirling cycle refrigerator (No. 35). The currently available machines have MTBF's that are not nearly high enough to fulfill the requirements of the FLIR. With additional development, however, a machine could be built that would be competitive with the VM refrigerator. The developed machine would probably employ a rhombic drive to decrease vibration and rolling metal seals to insulate the crankcase from the helium working volume, allowing oil lubrication of the crankcase. Some preliminary development work has shown this approach is feasible.

11.1 Recommendations For Additional Work

Depending on the procurement time schedule, additional development work on refrigeration systems for FLIR units may be attractive. If final decisions can be reached for firm operational requirements of the refrigerator, a split unit, if a split unit with oil lubricated compressor still fulfills the physical requirements, should be developed which exactly matches the operational requirements. Although the split units did rate high in the current evaluation, they did not exactly match the requirements of refrigeration capacity as a function of temperature, cooldown time, etc. This is reflected in the lower than perfect score for the units in these categories. The development could insure a solution to the compressor orientation problem and with some design effort, smaller compressors and cold heads may be possible. The development cost for this type of system would be relatively low because of the advanced state of the art for these refrigerators.

If the open cycle J-T system is acceptable as the primary refrigerator, tests to determine the long life capability of air purifiers should be conducted. Again, the tests should nearly duplicate the projected operating environment and duty cycle requirements of the cryogenic refrigerator with several J-T systems operating on high pressure air for extended periods of time. The operating experience gained from these tests would be valuable not only to show the overall reliability of the system but also to obtain experience on how and when purifier materials should be changed.

Development of an integral refrigerator to exactly match the FLIR operational requirements could also be attractive but the major need, especially for the VM refrigerator, is for life tests to determine a meaningful MTBF of a refrigerator designed for the FLIR use. The life tests should be designed to duplicate the environmental conditions anticipated and also the anticipated duty cycle, i. e., 12 hours on,

12 hours off. Since life tests of any significance must be over a longer period than necessary to have most of a large number of the machines under test to fail, a life test to prove the reliability of a machine should be initiated a year or longer before final installation of the refrigerator is anticipated.

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The Naval Ordnance Laboratory (NOL) has asked the Cryogenics Division of the National Bureau of Standards to investigate and evaluate 1) commercially available refrigerators, 2) refrigerators under development, and 3) new or novel ideas applicable to a refrigerator to cool infrared detectors in a shipboard Forward Looking Infrared (FLIR) system. Although a refrigerator has been selected for two prototype FLIR units, the study was initiated to select the most appropriate refrigerator for additional purchases of FLIR units. The FLIR requires a refrigerator capacity of approximately 2 watts at 77 K and a physical configuration allowing for an interface to the FLIR unit. Information was collected by interviewing FLIR manufacturers, surveying refrigerator manufacturers and by contacting users of similar systems. Correlation of this information with the NOL requirements is presented herein. The primary difference between airborne/spaceborne refrigerators and a shipborne refrigerator is the accessibility for minor repairs on-board ship although major repairs may be deferred for extended periods of time. It is anticipated that the shipboard units will operate away from major maintenance facilities, for periods as long as 6 months, with the refrigerator operating at least half of this time. Thus, reliability and ease of maintenance are emphasized when evaluating the various systems.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)*

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